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THERMAL ENGINEERING

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Pearson

Dedicated to my Parents

—Sadhu Singh

Dedicated to my beloved Parents
Late Sakti Pada Pati and Usha Rani Pati

—Sukumar Pati

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Preface

It gives us immense pleasure to present the book on *Thermal Engineering*. This book is intended for undergraduate students of mechanical, chemical or process, power, aerospace and automobile engineering. The content covers the syllabi of all the subjects pertaining to thermal engineering such as applied thermodynamics, power plant engineering, energy conversion and management, internal combustion engines, turbomachinery, gas turbines and jet propulsion, and refrigeration and air-conditioning taught at different levels of engineering curriculum.

From the vast experience of teaching, we feel that the students are lacking in fundamental concepts of different aspects of thermal engineering. The basic objective of an introductory course of thermal engineering should be to build fundamental understanding together with analytical and problem solving abilities. It is, therefore, imperative to accentuate the basic issues to develop a solid foundation during the different time-frame starting from the first year to the final year of undergraduate studies. The present text is an endeavour towards the goal by offering the students a text book on thermal engineering in a most comprehensive manner.

Salient features of the present text are given below:

- Detailed description of intricate physics behind every physical phenomenon
- Stresses on physical understanding to develop mathematical modelling skills for solving real-life problems related to thermal engineering
- After completion of illustrations, summaries are highlighted for quick revision of the topics and findings and conclusion are made therein
- A large number of Solved Examples, Multiple-choice Questions, Review Questions and Exercises in each chapter for easy assimilation of the subject matter
- Solved examples based on questions from reputed institutes/ universities as well as from Previous Question Papers of GATE/IES/ IAS
- Multi-choice Questions from GATE/IES/IAS with explanatory notes to help the students to prepare for different competitive examinations

This book provides a single core level text on ‘Thermal Engineering’. An effort has been made for a well-balanced coverage of physical concepts, mathematical operations and practical demonstrations within the scope of the course. Throughout the text, SI units have been used. This book is divided into twenty chapters.

Chapter 1 starts with the description of fuels and the detailed analysis of combustion. The analysis of flue gas and procedure for determination of calorific value of fuel using different types of calorimeter are discussed. *Chapter 2* deals with properties of steam. The property changes during transformation of steam into different phase and for different processes are shown through different plots like p–v, T–s and h–s. Dryness fraction of steam and its measurement using different calorimeter are discussed. *Chapter 3* discusses different types of steam generators used for power generation. *Chapter 4* deals with steam power cycle where basic and improved cycles are discussed. *Chapter 5* is devoted to describe the working of

steam engine together with the mathematical analysis. An elaborate in-depth discussion on flow through steam nozzle is presented in *Chapter 6*.

The next two chapters (*Chapters 7* and *8*) deal with the principle of operation of different types of steam turbines and steam condensers. *Chapter 9* discusses the gas power cycles utilised in various cycles. The working principle and the function of different components of internal combustion engines are discussed at length in *Chapter 10*. Two stroke and four stroke engines as well as spark ignition and compression ignition engines are considered. *Chapter 11* deals with the analysis of performance characteristics of internal combustion

engines at different working conditions.

The next four chapters (*Chapters 12*, *13*, *14 and 15*) deal with the working principle of four different types of compressors, namely, reciprocating, rotary, centrifugal and axial flow compressors. *Chapter 16* is concerned with the details of different components of gas turbine and their functions. *Chapter 17* is devoted to present different jet propulsion systems together with the associated cycles. Different refrigeration systems and their components are discussed at length in *Chapters 18 and 19*. *Chapter 20* deals with psychrometrics where the properties of atmospheric air together with different processes are presented by

psychrometric chart. Fundamentals of different air-conditioning systems are discussed.

Apart from undergraduate students, this book will provide fundamental information as required by the practising engineers in the field actual operation in different industries. Students pursuing diploma can also be benefited out of this text. The text would be extremely helpful for various competitive examinations such as GATE, IES, IAS, etc.

The complete solutions manual is also available for the faculty members at www.pearsoned.co.in/sadhusingh

Prof. Singh places on record with

gratitude the moral support received from his family members, children and grandchildren. Dr Pati is indebted to his colleagues and students at NIT Silchar for many fruitful interactions. Dr Pati expresses his gratitude to his mother, father-in-law, mother-in-law, wife Munmun, son Soumik and relatives for their persistent encouragement and support and the tolerance they have meted out during long absence from family life while perusing the present endeavour. The cooperation received from the publishers and in particular from Ms. Harsha Singh and Mr. M. Balakrishnan is highly appreciated.

Despite all our efforts, it is possible that some unintentional errors are still

remaining in the text. We would gracefully acknowledge if any such errors are pointed out for necessary corrections and would be extremely thankful to the readers for their constructive suggestions and healthy criticism for further improvement of the text. If you have comments, suggestions or corrections, please email us at sukumarpati@gmail.com.

Sadhu Singh
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About the Authors

Late Dr Sadhu Singh retired as Director (Colleges), Punjab Technical University, Jalandhar. He had attained his B.Sc. Mechanical Engineering (Honours) from Punjab University, Chandigarh, M.Sc. (Mechanical Design and Production Engineering) and Ph.D. from Kurukshetra University. His teaching experience spans 15 years at Regional Engineering College, Kurukshetra (now NIT Kurukshetra) and 19 years at Pantnagar University. He had been Professor and Head of Mechanical Engineering Department and Dean, Faculty of Engineering and Technology at G. B. Pant University of

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Sukumar Patis currently working as Assistant Professor in the Department of Mechanical Engineering at National Institute of Technology (NIT) Silchar, India. Prior to joining at NIT Silchar, he worked as a Post-doctoral Fellow for almost one year at Indian Institute of Technology Kharagpur after completion of his Ph.D. from Mechanical Engineering Department of the same Institute. He was awarded a medal for attaining the second rank in B.E. in Mechanical Engineering from the University of North Bengal. He has a teaching and research experience of more than seventeen years.

Dr Pati has authored a textbook titled *A*

Textbook on Fluid Mechanics and Hydraulic Machines, and is the co-author of the textbooks like *Engineering Mechanics and Engineering Thermodynamics and Fluid Mechanics*. He has published several research papers in peer-reviewed international journals, including papers in top-ranked prestigious journals such as *International Journal of Thermal Sciences, International Journal of Heat and Mass Transfer, Physical Review E, International Communications in Heat and Mass Transfer, Energy, ASME Journal of Heat Transfer, Numerical Heat Transfer, Part A: Applications, Nanoscale and Microscale Thermophysical Engineering and Proceedings of the Institution of*

Mechanical Engineers, Part E: Journal of Process Mechanical Engineering. His field of expertise is thermal science and engineering, and in particular, evaporation and condensation, micro-scale fluid flow and heat transfer, natural and mixed convection, computational fluid dynamics and non-Newtonian fluid mechanics.

Chapter 1

Fuels and Combustion

1.1 ❒ INTRODUCTION

A fuel may be defined as a substance, which on burning with oxygen produces a large amount of heat. A fuel is mainly composed of carbon and hydrogen. The energy produced by burning of fuel in the form of heat is known as chemical energy.

1.2 ❒ CLASSIFICATION OF FUELS

Fuels may be classified as follows:

1. Based on their occurrence, they are classified as natural fuels and artificial (or prepared) fuels.

   Natural fuels occur in nature and they are also called primary fuels, whereas artificial fuels are

prepared by further processing of primary fuels and they are also called secondary fuels.

2. Based on their state, they are classified as solid fuels, liquid fuels, and gaseous fuels.

A detailed classification of fuels is given in Table 1.1.

Table 1.1 *Classification of fuels*

1.3 ❐ SOLID FUELS

1.3.1 Primary Fuels

1. **Wood:** It is the most commonly used and easily available natural fuel. It has been used for cooking and other purposes over centuries. The important constituents of wood are cellulose fibre and water. On burning, it gives ash, which is used for cleaning utensils in rural and even urban areas. The calorific value of wood varies with its type and moisture content. It ignites easily at about 250°C and is used for igniting other fuels.
2. **Peat:** It is a mixture of water and decayed vegetable matter. It contains a large amount of moisture and requires drying in sun for about one to two months before use. It is the first stage in the formation of coal from wood.
3. **Lignite:** It is transformed from peat. It is brown in colour and is also known as brown coal. Air-dried lignite contains 10%–20 % moisture. It is used as low-grade fuel. It is non–caking

and burns with a large smoky flame.

4. **Bituminous coal:** It is transformed from lignite. It has a shining, black appearance and comes in caking and non–caking varieties. It ignites easily and burns with a long yellow and smoky flame.
5. **Anthracite coal:** It is transformed from bituminous coal. It is hard, brittle, and lustrous in appearance. It is non–caking and burns without smell, smoke, or flame. It is difficult to ignite. This coal has minimum ash, volatile matter, and moisture. Its calorific value is the highest and is suitable for steam generation in thermal power plants.

The calorific value of solid fuels is given in Table 1.2.

Table 1.2 *Calorific value of solid fuels*

1.3.2 Secondary Fuels

1. **Wood charcoal:** It is obtained by destructive distillation of wood in a crude method. It is prepared by slow burning of logs in a domed type of earthen structure under controlled conditions of air supply. It takes many days to obtain charcoal. During the process, the volatile matter and water are expelled to the atmosphere. The temperature of heated air is not allowed to increase beyond about 300°C. It burns rapidly with a clear flame without smoke. It is soft and dark black in colour. It is mainly used for domestic purposes.
2. **Coke:** It is prepared by removing the volatile matter from bituminous coal. It is hard, brittle, and porous. It is generally prepared by heating in an electric furnace, followed by water sprays. It contains carbon, 2% Sulphur, and small quantities of hydrogen, nitrogen, and phosphorus. It is mainly used in blast

furnace to produce heat and reduce iron ore.

3. **Briquetted coal:** It consists of finely ground coal or coke mixed with a suitable binder and pressed together to form blocks or briquettes of any shape. In the briquetted form, the heating value of low-grade coal is increased.
4. **Pulverised Coal:** The crushed coal to powder is called pulverised coal. The fineness of powdered coal is so adjusted that it floats during the burning process. This gives better contact between air and fuel, which results in very high combustion efficiency. The pulverising method is used for steam raising in boilers by using low grade and rough fuels. The advantages of coal pulverising are as follows:
    1. Flexibility of control
    2. Complete combustion with lesser excess air
    3. High flame temperature

1.3.3 Desirable Properties of Coal

A good coal should have the following properties:

1. Low ash content
2. High calorific value
3. Small percentage of sulphur (<1%)
4. Good burning characteristics, that is, it should burn freely without agitation
5. High grindability index for pulverising purposes
6. High weatherability

1.3.4 Ranking of Coal

Rank is an inherent property of the fuel, depending upon its relative progression in the classification process. The ranking

of coal is based on the fixed carbon and heating value of the mineral matter free analysis. The ASME (American Society of Mechanical Engineers) and ASTM (American Society of Testing and Materials) have accepted the following ranking specifications for coal:

1. **Higher ranking:** It is done on the basis of heating value (moist basis).
2. **Lower ranking:** It is done on the basis of heating value (most basis).

For example, A coal ranked as 62–20,000 means that the coal contains C = 62% and C.V. = 20,000 kJ/kg.

1.3.5 Grading of Coal

The grading of coal is done on the following basis:

1. Size
2. Heating value
3. Ash content

4. Ash softening temperature
5. Sulphur content

For example, a coal grade written as 5–10 cm, 20,000 – A8-F24 – F1.6 indicates that the coal has

Size: 5–10 cm

C.V.: 20,000 kJ/kg

Ash: 8–10%

Ash softening temperature: 2400–2500°F

Sulphur content: 1.4–1.6%

1.4 ❐ LIQUID FUELS

The main source of liquid fuels is petroleum which is obtained from drilled wells under the Earth's crust.

Petroleum has originated probably from organic matter such as fish and plant either by bacterial action or distillation under pressure and heat. It consists of a mixture of gases, liquids, and solid hydrocarbons with small amounts of nitrogen and sulphur compounds.

Liquid fuels are classified as follows:

1. **Natural or crude oils:** These oils are further distilled to obtain petrol, benzene, diesel, and other light oils such as kerosene, alcohol, and so on.
2. **Distilled artificial oils:** These oils include coal–tar, tar–oil, shale–oil, and natural gas oil.

Various oils obtained from natural or crude oils are briefed below.

1. **Petrol:** It is the lightest and the most volatile liquid fuel. It is mainly used for spark ignition petrol engines. It is obtained by fractional distillation of petroleum at 65°C to 220°C. It is also known as gasoline.
2. **Kerosene:** It is obtained by fractional distillation of crude oil at 345°C to 365°C. It is also known as paraffin oil. It is heavier and less volatile than petrol. It is used for heating and lighting purposes.
3. **Diesel:** It is obtained by fractional distillation of crude oil at

345°C to 470°C. It is mainly used in diesel engines and oil–fired furnaces and boilers. It is also known as heavy fuel oil.

4. **Benzene:** It is obtained by the redistillation of tar, which is the byproduct collected during the production of coal gas. It is used as an alternate fuel for internal combustion (IC) engines. It is less prone to detonation than standard petrol.
5. **Alcohol:** It is formed by the fermentation of vegetable matter. The calorific value of alcohol is lower than that of petrol and the cost is higher. It is mainly used for industrial purposes.

1.4.1 Advantages and Disadvantages of Liquid Fuels Over Solid Fuels

The advantages of liquid fuels over solid fuels are as follows:

1. Requirement of less space for storage
2. Higher calorific value
3. Easy handling and transportation
4. Better control of consumption by using valves and number of burners
5. Absence of danger from spontaneous combustion
6. Better cleanliness and absence from dust
7. Absence of ash problems
8. No problem of corrosion and deterioration during storage
9. Higher combustion efficiency

The disadvantages of liquid fuels over solid fuels are as follows:

1. Higher cost
2. Greater risk of fire
3. Require costly containers for storage and transportation

1.4.2 Calorific Value of Liquid Fuels

The calorific value of liquid fuels is given in Table 1.3.

Table 1.3 *Calorific value of liquid fuels*

1.4.3 Desirable Properties of Liquid Fuels

Liquid fuels should have the following desirable properties:

1. Low ash content
2. High calorific value
3. Low gum content
4. Less corrosive tendency
5. Low sulphur content
6. Low pour point

1.5 ❐ GASEOUS FUELS

1. **Natural gas:** The main constituents of natural gas are methane (CH_4) and ethane (C_2H_6). Its calorific value is nearly 21 MJ/m^3. It is used as an alternate fuel for internal combustion engines as liquefied natural gas (LNG).
2. **Coal gas:** It mainly consists of hydrogen, carbon monoxide, and hydrocarbons. It is prepared by carbonisation of coal. It is also known as town gas. It is largely used in towns for street and domestic lighting, cooking, and for running gas engines.
3. **Coke–oven gas:** It is obtained during the production of coke

by heating bituminous coal. The volatile content of coal is driven off by heating. It is produced by the destructive distillation of coal in air–tight coke ovens made of silica bricks. It is used in heating ovens. This gas must be thoroughly filtered before using in gas engines.

4. **Blast furnace gas:** It is obtained as a byproduct from the smelting operation in which air is forced through layers of coke and iron during the manufacture of pig iron. It contains about 20% carbon monoxide. The gas leaving the blast furnace has a high dust content, which must be removed by dust collectors and washing operation. It has a low calorific value. It is mainly used as a fuel for metallurgical furnaces.
5. **Producer gas:** It is obtained from the partial oxidation of coal, coke, or peat when burnt with an insufficient quantity of air. It has low heating value and in general is suitable for large installations. It is also used in steel industry for firing open hearth furnaces.
6. **Water gas:** It is produced by blowing steam into white hot coke or coal. It is a mixture of hydrogen and carbon monoxide. It burns with a blue flame and is also known as blue water gas. It does not contain unsaturated hydrocarbons. In order to use this gas for domestic lighting, unsaturated hydrocarbons are added. In such cases, it is also known as carburetted water gas or enriched water gas. This is achieved by passing the gas through a carburettor into which a gas oil is sprayed. It is usually mixed with coal gas to form town gas. It is used in furnaces and for welding.
7. **Sewer gas:** It is obtained from sewage disposal vats in which fermentation and decay occur. It consists of mainly marsh gas (CH_4) and is collected at large disposal plants. It is used as a fuel for gas engines which, in turn, drives plant pumps and agitators.

1.5.1 Calorific Value of Gaseous Fuels

The calorific value of gaseous fuels is given in Table 1.4.

Table 1.4 *Calorific value of gaseous fuels*

1.5.2 Advantages and Disadvantages of Gaseous Fuels

The advantages of gaseous fuels are as follows:

1. Better control of combustion
2. Excess air required for complete combustion is very less
3. Can be directly used in IC engines
4. Give higher combustion efficiency
5. Free from solid and liquid impurities.
6. Do not produce ash and smoke.
7. No problem of shortage if supply is available from public supply line.
8. Distribution of gaseous fuels even over a wide area is easy with the help of pipe line.

The disadvantages of gaseous fuels are as follows:

1. They are highly inflammable.
2. The storage space required is huge if public supply line is not available.

1.5.3 Important Properties of Gaseous Fuels

The important properties of gaseous fuels are as follows:

1. Calorific value
2. Viscosity
3. Specific gravity
4. Density
5. Diffusibility

1.6 ❒ LIQUEFIED GASES

Petroleum gases can be liquefied by slightly compressing them. When these gases liquefy, they contract and occupy less space. They can then be transported more easily in liquid form in pressurised containers. If the pressure is released by opening the valve, the liquid from the container begins to evaporate and provides a continuous supply of gas.

There are only two families of

petroleum gases which can be liquefied by compression at atmospheric pressure. These are as follows:

1. Propanes (C_3H_8) and Propylenes (C_3H_6)
2. Butanes (C_4H_{10}) and Butylenes (C_4H_8)

Propane and butane are paraffin hydrocarbons and have the general formula as C_nH_{2n+2}. Methane (CH_4) and ethane (C_2H_6) must be chilled before they can be liquefied.

Another series of hydrocarbons associated with the aforementioned paraffin is called Olefins and have the general formula as C_nH_{2n}. These include ethylene, propylene, butylene, etc.

1.6.1 Liquefied Petroleum Gas

Liquefied petroleum gas (LPG) mainly

contains propane and some fractions of butane. It must be stored under pressure to keep it in liquid form. LPG is gaining popularity as a gasoline substitute in IC engines because it costs 60% of petrol and gives 90% mileage of its fellow gasoline. It burns more clearly, has a higher octane number. Large quantities of LPG are now available from gas and petroleum industries. They are often used as fuel for tractors, trucks, and as domestic fuel for cooking. The calorific value of LPG is about 45,360 kJ/kg.

1.6.2 Liquefied or Compressed Natural Gas

LNG comes from dry natural reservoirs, mainly constituting about 85–90% of methane (CH_4) with very small percentages of ethane and propane. Its

critical temperature is about 73°C. It is found in coal reserves in tight sands and is trapped in geo-pressurised zones within the earth. LNG is obtained when natural gas (NG) is placed under very high pressure. On the release of high pressure, LNG converts to NG. At atmospheric pressure and above freezing temperature, LPG is in gas form.

Due to the low critical temperature of NG, it cannot be liquefied at ordinary temperature. To liquefy NG, it must be cooled to cryogenic (ultra-low) temperatures. It must be stored in well-insulated containers so as to hold it in liquid form. The pressure must be maintained at about 50–60 bar. For automobiles, a pressure of 220–250 bar

is required. The NG used in automobiles is known as CNG. NG used for domestic and industrial purposes is supplied in pipes and is hence known as piped natural gas (PNG). The calorific value of LNG is 40.7–41.2 MJ/m^3.

1.7 ❐ BIOFUELS

The organic material produced by plants and their derivatives is called biomass. It includes forest crops and residues, energy crops, and animal manure. It reacts with oxygen in combustion and natural metabolic process to release heat. The material may be transformed by chemical and biological processes to produce intermediate biofuels such as methane gas, ethanol, or charcoal solid. Sugarcane, poplar, eucalyptus, animal,

and human waste include biomass which are used as biofuels.

Biofuel production is economical if the production process uses materials as by-products. They are available at low cost. Some examples are waste from animal enclosures, offcuts and trimmings from sawmills, municipal sewage, rice husk and straw, wheat straw, shells from coconuts, and straw from cereal crops. The main dangers of extensive use of biomass fuel are deforestation, soil erosion, and displacement of food crops by fuel crops.

1.8 ❐ ANALYSIS OF FUELS

The analysis of solid fuels may be carried out in two ways: proximate analysis and ultimate analysis.

1.8.1 Proximate Analysis

The proximate analysis of coal is carried out to determine its behaviour when heated. It gives the percentage of fixed carbon, volatile matter, moisture content, and ash content. This analysis is sufficient for commercial purposes.

The following procedures are adopted to estimate the various contents of coal:

1. **Moisture content:** One gram sample of coal is heated to a temperature of about 105°C for a period of one hour. The loss in weight gives the moisture content of the coal.
2. **Volatile matter:** One gram sample of coal is placed in a covered platinum crucible and heated to 950°C. This temperature is maintained for about seven minutes. There is a loss in weight due to vapourisation of moisture and volatile matter. Since the moisture content has already been calculated, the volatile matter can be obtained as

   Volatile matter = total loss in weight – moisture content

3. **Ash content:** One gram sample of coal in an uncovered crucible is heated to a temperature of about 720°C until the coal is completely burned. Complete combustion of coal is ascertained by repeated weighing of sample. This ensures that there is only ash remaining in the crucible.
4. **Fixed carbon:** Fixed carbon can be obtained as

Fixed carbon = (100% by mass of coal) – (moisture content + volatile matter + ash)
However, this difference does not represent all the carbon present in the coal, because some of the carbon in the form of hydrocarbons may have been lost while estimating the volatile matter.

1.8.2 Ultimate Analysis

The ultimate analysis gives the percentage of each chemical element in coal, along with ash and moisture. This analysis gives the following components on mass basis:

Carbon (C), hydrogen (H_2), oxygen (O_2), nitrogen (N_2), sulphur (S), moisture (M), and ash (A).

Thus, $C + H_2 + O_2 + N_2 + S + M + A = 100\ \%$ by mass.

1.9 ❒ CALORIFIC VALUE OF FUELS

The calorific value of a solid or liquid fuel is defined as the heat evolved by the complete combustion of unit mass of fuel. The calorific value of a gaseous fuel is expressed as the heat evolved by the complete combustion of one cubic metre of gas at standard temperature and pressure. There are two calorific values of fuels.

Higher Calorific Value

All fuels containing hydrogen produce water vapour during combustion. If these products of combustion are cooled to the room temperature, the water vapour will condense evolving its latent heat of vapourisation, producing the maximum amount of heat per kg of fuel. This heat is known as higher calorific

value (HCV).

Lower Calorific Value

The cooling of flue gases to room temperature is not possible in most combustion processes. Therefore, the amount of latent heat of water vapour goes waste. Accordingly, for the calculation of the lower calorific value, we assume that the water vapour formed during combustion leaves as vapour.

LCV = HCV – heat carried away by water vapour formed per kg of fuel burned.

where m_w = mass of water vapour formed per kg fuel burned, h_{fg} = latent heat of vapourisation at the partial pressure of water vapour in the

combustion products = 2395 kJ/kg.

The HCV of fuel is given by

where C, H, O, and S are percentages of carbon, hydrogen, oxygen, and sulphur, respectively.

The LCV of fuel is given by

1.10 ❐ COMBUSTION OF FUELS

The following chemical equations are used to calculate the amount of oxygen required and the amount of gases produced by the combustion of fuel, by using the molecular weight of the elements in kg:

1. $C + O_2 \rightarrow CO_2$

   $12 + 32 \rightarrow 44$

2. $2C + O_2 \rightarrow 2CO$

$24 + 32 \rightarrow 56$

3. $2CO + O_2 \rightarrow 2CO_2$

$56 + 32 \rightarrow 88$

4. $2H_2 + O_2 \rightarrow 2H_2O$

$4 + 32 \rightarrow 36$

$1 + 8 \rightarrow 9$

5. $S + O_2 \rightarrow SO_2$

$32 + 32 \rightarrow 64$

$1 + 1 \rightarrow 2$

1.11 ❐ COMBUSTION OF HYDROCARBON FUEL

For any hydrocarbon fuel, we have

$$xC_nH_{2n} + 2 + yO_2 \rightarrow aCO_2 + bH_2O$$

where x, y, a, and b are constants to be evaluated.

Equating the weights of C, H, and O on both sides of the equation, we have

$$xn = a$$

or $a = xn$

$$x(2n + 2) = 2b$$

or $b = [x/2]\,(2n + 2) = x(n + 1)$

$$2y = 2a + b$$

or $y = a + [b/2] = xn + [x/2]\,(n + 1) = [x/2]\,(3n + 1)$

Substituting there values in combustion equation, we get

1.12 ❐ MINIMUM AIR REQUIRED FOR COMPLETE COMBUSTION OF SOLID/LIQUID FUELS

Let C, H, O, and S = percent by mass of C, H_2, O_2, and S in fuel. Then, the mass of O_2 required per kg of fuel for

complete combustion

Percentage of O_2 by mass in air = 23%

∴ Minimum amount of air required in kg per kg of fuel for complete combustion

1.13 ❐ CONVERSION OF VOLUMETRIC ANALYSIS TO MASS (OR GRAVIMETRIC) ANALYSIS AND VICE-VERSA

Let the fraction of CO, CO_2, N_2, and O_2 by volume of dry exhaust gases by C_1, C_2, N, and O, respectively, obtained by the Orsat apparatus. The volumetric analysis can be converted into gravimetric analysis, as shown is Table 1.5.

Table 1.5 *Conversion from volumetric to gravimetric analysis*

$$\Sigma(4) = \Sigma X = 28\ C_1 + 44\ C_2 + 28\ N + 32\ O$$

The conversion from gravimetric to volumetric analysis is given in Table 1.6.

Table 1.6 *Conversion of gravimetric analysis to volumetric analysis*

1.14 ❒ DETERMINATION OF AIR SUPPLIED

1.14.1 Percentage of Carbon by Mass in Fuel and Volumetric Analysis is Known

We know that 1 kg of flue gas contains kg of CO and kg of CO_2

Amount of carbon in kg of

Amount of carbon in kg of

Total mass of carbon in 1 kg of flue gas kg

Let C = mass fraction of carbon in fuel and there is no nitrogen in fuel.

m_g = mass of flue gases formed per kg of fuel burned.

Then,

Now, nitrogen in 1 kg of air supplied = 0.77 kg

N_2 in 1 kg of flue gas =

∴ Air supplied per kg of flue gas =

Mass of air supplied per kg of fuel = mass of air supplied per kg of flue gas ×

mass of flue gases formed per kg of fuel.

1.14.2 Excess Air Supplied

The oxygen carried in the flue gases comes only from the air supplied for the combustion of fuel. In addition, the oxygen necessary to convert CO to CO_2 should be taken into account in calculating the excess air supplied.

CO in 1 kg of flue gas =

O_2 in 1 kg of flue gas =

O_2 required for burning kg of CO to CO_2 =

Excess O_2 available per kg of flue gas formed

Excess O_2 supplied per kg of fuel = Excess O_2 per kg of flue gas × m_g

Excess air supplied per kg of fuel

1.15 ❐ DETERMINATION OF PERCENTAGE OF CARBON IN FUEL BURNING TO CO AND CO_2

1. Let the volumetric composition of dry flue gases be known.

   Percentage of C burning to CO_2

   Similarly, percentage of C burning to CO

2. When gravimetric analysis is known, then

1.16 ❐ DETERMINATION OF MINIMUM QUANTITY OF AIR REQUIRED FOR COMPLETE COMBUSTION OF GASEOUS FUEL

Let the volumetric analysis of gaseous fuel be known. The chemical equations for gaseous fuel are as follows:

1. $2H_2 + O_2 \rightarrow 2H_2O$

2 volumes of H_2 + 1 volume of $O_2 \rightarrow$ 2 volumes of H_2O

or 1 volume of H_2 + volume of $O_2 \rightarrow$ 1 volume of H_2O

2. $2CO + O_2 \rightarrow 2CO_2$

   2 volumes of CO + 1 volume of $O_2 \rightarrow$ 2 volumes of CO_2

3. $CH_4 + 2O_2 \rightarrow 2H_2O + CO_2$

   1 volume of CH_4 + 2 volumes of $O_2 \rightarrow$ 2 volumes of H_2O + 1 volume of CO_2

4. $C_2H_4 + 3O_2 \rightarrow 2H_2O + 2CO_2$

   1 volume of C_2H_4 + 3 volumes of $O_2 \rightarrow$ 2 volumes of H_2O + 2 volumes of CO_2

Let the volume fraction of the gaseous fuel contain H_2, CO, CH_4, C_2H_4, CO_2, and N_2, the last two being incombustible gases.

The quantity of O_2 required m^3/m^3 of gas

Air contains 21% of O_2 by volume.

Quantity of air required

1.17 ❐ DETERMINATION OF EXCESS AIR SUPPLIED FOR GASEOUS FUEL

Let the volumetric fractions of gaseous fuel and dry flue gases be known.

Let V = volume of flue gases formed in m^3 per m^3 of fuel when air supplied is just sufficient for complete combustion.

V_1 = volume of excess air supplied in m^3 per m^3 of fuel gas.

Total volume of dry flue gases formed per m^3 of gaseous fuel = $V + V_1$

Volume of excess air in flue gases per

m^3 of flue gas m^3 of air/m^3 burned gases.

Let O = quantity of O_2 in m^3/m^3 of exhaust gas

Then air in the exhaust gas = m^3 of air/m^3 of flue gases

1.18 ❒ FLUE GAS ANALYSIS

1.18.1 Orsat Apparatus Construction

The construction of the Orsat apparatus is shown in Fig. 1.1. It consists of three flasks *a*, *b,* and *c* each containing different chemicals for absorbing CO_2, O_2, and CO. The percentage of N_2 is obtained by difference. The percentage of SO_2 in the gases cannot be measured by this apparatus. The absorbents used

in flask *a* are NaOH or KOH solution to absorb CO_2. The tube *b* contains alkaline solution of pyrogallic acid to absorb O_2. The flask *c* contains cuprous chloride to absorb CO. Keeping values *X*, *Y*, and *Z* closed, the three-way valve *A* is opened and the aspirator bottle *B* is moved down so that the flue gas enters the eudiometer tube. The flue gas is drawn in until the level in the eudiometer reads zero. The three-way valve *A* is closed and then the value *X* of the flask *a* is opened and the aspirator bottle is moved up and down several times pushing the gas into the flask *a* containing KOH solution to absorb CO_2 gas. Now, the flue gas is taken into the eudiometer by lowering the aspirator bottle and the valve *X* is closed, keeping

the original level of the solution. The aspirator is then brought near the eudiometer and placed at such a position that the water level in both places is the same and the reading of the eudiometer tube is taken. The difference in reading gives the percentage of CO_2 in the flue gas. The procedure is repeated with the flasks *b* and *c* to find the percentage of O_2 and CO. The remainder of the flue gas is taken as N_2.

Figure 1.1 *Orsat apparatus*

This apparatus gives the percentages of dry flue gases only, as water vapour is condensed at room temperature when the gas is drawn over water. The flue gases may be passed through a *U*-tube containing calcium chloride before

passing into the Orsat apparatus to ensure that only dry gas is being analysed.

Test Procedure

Before starting the experiment it is essential that the chemical solutions in the flask are fresh and free from pollution. Then, water is filled in the eudiometer jacket and salt water in the aspirator bottle. To expel the residual gases, if any, from the eudiometer, the three-way valve is opened to the atmosphere and the aspirator bottle is raised to fill the eudiometer to the 100 cc mark. The three-way valve is then closed and the valve x is opened. The aspirator bottle is lowered which, in turn, lowers the level of salt water in the

eudiometer resulting in the rise in the level of chemical solution in flask *a*. As soon as the chemical solution touches a mark on the capillary connecting the common header tube, the valve *x* is closed. The same procedure is repeated for flasks *b* and *c*. By doing so, all the residual gases in the flasks are expelled to the eudiometer and the common header. The residual gases from the eudiometer are expelled by opening the single way valve *C* to the atmosphere and raising the aspirator bottle so that salt water reaches up to the 100 cc mark. The apparatus is now ready for taking readings of flue gases.

1.19 ❐ BOMB CALORIMETER

1.19.1 Construction

This is used to determine the higher calorific value of solid and liquid fuels. The bomb calorimeter shown in Fig. 1.2 consists of a strong steel shell, called the bomb, which can withstand a pressure of about 100 atm. The electric supply is provided at the bottom of the bomb. The silica or quartz crucible is supported on two pillars. It is also provided with a non-return and release valves. The bomb is placed in water bath and the water bath itself is placed in another container which acts as heat insulator. A thermometer and a stirrer are inserted in the outer vessel.

1.19.2 Working

A known quantity of fuel is taken in the crucible such that the fuse touches the

fuel. The bomb is then charged with oxygen to a pressure of 30 bar. The oxygen charging non-return valve and release valve are closed tightly and the bomb is placed in water bath and closed with the cover. The water in the calorimeter is stirred and when the temperature remains steady, the fuel is ignited by passing a current through the fuse wire. The temperature of the bath starts increasing very quickly after the ignition and the readings on the thermometer are taken at regular intervals till the maximum temperature is reached.

Figure 1.2 *Bomb calorimeter*

The temperature gradually starts falling. When the temperature fall shows a

steady rate, the readings are taken again for another five minutes to determine the cooling correction, as shown in Fig. 1.3. The cooling correction should be added to the measured temperature rise.

Figure 1.3 *Temperature-time curve*

Considering heat balance, we have

Heat given by the fuel due to combustion + Heat given by the combustion of fuse wire = Heat absorbed by the water and calorimeter.

$$m_f \times \text{HCV} + m_{fw} \times \text{C.V.} = (m_w + m_a)\Delta\theta c_p$$

where m_f = mass of fuel

m_{fw} = mass of fuse wire

m_c = water equivalent of calorimeter

m_w = mass of water in calorimeter

c_p = specific heat of water

θ_m = recorded temperature rise

HCV = higher calorific value of fuel

C.V. = calorific value of fuse wire

$\Delta\theta$ = true temperature rise

The water equivalent of calorimeter is determined by burning a fuel of known calorific value and using the aforementioned equation. The fuels used for this purpose are benzoic acid (C.V. = 111573 kJ/kg) and naphthalene (C.V. =

40690 kJ/kg).

1.19.3 Cooling Correction

The radiation correction is computed from Newton's law of cooling. It states that the rate of heat loss due to radiation is proportional to temperature difference between the hot body and the surroundings. Thus,

$$\delta Q \propto (\theta_2 - \theta_1)dt$$

The rate of temperature drop is determined from the temperature-time graph for the bomb calorimeter. Let it be r°C after the maximum temperature is reached. Therefore, during the period the temperature rises, it can be assumed that fall in temperature occurs due to

radiation. If t is the time required for the maximum temperature to be reached, the temperature fall during that period due to radiation is Thus, for temperature fall,

1.20 ❐ BOYS GAS CALORIMETER

1.20.1 Construction

It is used to determine the higher calorific value of gaseous fuels. Figure 1.4 shows the components of the calorimeter. It consists of a gas burner B with arrangements to measure flow quantity and pressure of gas supplied to burner. A gas pressure regulator is used in the gas supply line to control gas flow and pressure fluctuation. The fuel burns within a cylindrical container surrounded by a cooling coil. Cooling

water is supplied to the cooling coil from a constant heat tank. The outer casing of the calorimeter is provided with heavy insulation to prevent any heat loss to the surroundings.

Figure 1.4 *Boys gas calorimeter*

1.20.2 Working

The flue gas moves up the cylindrical container and flows down from the top of the container. The steam formed due to the combustion of hydrogen in the fuel and carried with the gases condenses around the cooling coil and drips down into the trap below. The overflow of the condensate is taken out in a glass beaker.

Calculations

Let V_g' = volume of gas consumed, m^3

m_w = mass of water circulated, kg

$(\Delta\theta)_w$ = rise in temperature of water circulated, °C

h_w = pressure of gas above atmosphere, cm of water

h_b = barometer reading, cm of Hg

θ_g = temperature of gas supply, °C

Volume of gas at normal temperature and pressure (NTP),

The calorific value of fuel is given by

$$V_g \times \text{HCV} = m_w \times (\Delta\theta)_w$$

Example 1.1

A coal has the following composition by mass:

$$C = 0.89,\ H_2 = 0.03,\ S = 0.01,\ O_2 = 0.02, \text{ and } N_2 = 0.03$$

and remaining is ash. Calculate the HCV and LCV of coal.

Solution

HCV = 35000 C + 143000 + 9160 S

= 35000 × 0.89 + 143000+ 9160 × 0.01

= 35174 kJ/kg

$$LCV = HCV - 9H \times 2460 = 35174 - 9 \times 0.03 \times 2460$$

$$= 34510 \text{ kJ/kg}$$

Example 1.2

The following observations were made to find the HCV and LCV of a solid fuel by using a bomb calorimeter:

Solution

Corrected temperature rise, $\Delta\theta_c = 2.97 + 0.03 = 3°C$

$$m_f \times \text{HCV} + m_{fw} \times \text{C.V.} = c_{pw}\,(m_w + m_c).\ \Delta\theta_c$$

or $0.8 \times 10^{-3} \times \text{HCV} + 0.02 \times 10^{-3} \times 6500 = 4.2\ (1.90 + 0.35) \times 3$

or HCV = 35275 kJ/kg

LCV = HCV – 9H × 2460 = 35275 – 9 × 0.04 × 2460

= 34389.4 kJ/kg

Example 1.3

The following observations were made by using a Boys gas calorimeter to determine the LCV of a gaseous fuel:

Gas burned = 50 litres

Gas pressure = 4.5 cm of water above atmospheric pressure

Barometer reading = 750 mm of Hg

Temperature of gas = 30°C

Water circulated through the calorimeter = 19 kg

Rise in temperature of water = 10°C

Condensate collected during the test = 60 gram.

Determine the HCV and LCV of the gas at 0°C and 760 mm of mercury pressure.

Solution

Volume at N.T.P., $V_g = V_g'$

$= 50 \times 10^{-3}$ 0.04465 m^3

HCV = = 17872 kJ/m^3 at N.T.P.

Condensate formed per m^3 of gas used at N.T.P.

LCV = HCV –1.1198 × 2460 = 15117 kJ/m^3

Example 1.4

The percentage composition by mass of a solid fuel is as follows: C = 90%, H_2 = 3.5%, O_2 = 3%, N_2 = 1%, S = 1%, and the remainder being ash. Find (a) the mass of air

required per kg of fuel of complete combustion, (b) the mass analysis of the dry products of combustion, (c) If 60% excess air is supplied in actual combustion, determine the mass of flue gases per kg of fuel, and (d) Volumetric analysis of the products per kg of fuel.

Solution

- Total O_2 required for complete combustion = $2.69 - 0.03 = 2.66$ kg/kg of fuel

  Air required = $2.66 \times = 11.58$ kg/kg of fuel

- Mass analysis of dry products of combustion:

$$N_2 = 11.58 \times 0.77 + 0.01 = 8.92 \text{ kg}$$

- For 60% excess air,

$$N_2 = 11.58 \times 1.6 \times 0.77 + 0.01 = 14.276 \text{ kg}$$

$$O_2 = 11.58 \times 0.6 \times 0.23 = 1.598 \text{ kg}$$

- Volumetric analysis

Example 1.5

The ultimate analysis of a coal is as follows:

$C = 82\%$, $H_2 = 5\%$, $O_2 = 8\%$, and the rest is ash.

The volumetric analysis of dry flue gases is:

$CO_2 = 10\%$, $CO = 2\%$, $O_2 = 8\%$, and $N_2 = 80\%$.

Find (a) air supplied per kg of fuel, (b) flue gases formed per kg of fuel, (c) percentage of excess air

supplied, (d) heat carried away by the products of combustion per kg of fuel if exit temperature of flue gases is 380°C, and (e) heat lost by incomplete combustion. Assume C.V. of carbon = 35000 kJ/kg when it burns to CO_2 and 10200 kJ/kg when it burns to CO. Air contains 23% of oxygen by mass. c_p for dry flue gas = 1 kJ/kgK and for steam, c_p = 2.10 kJ/kgK.

Solution

- Mass of carbon per kg flue gas = mass of C from CO_2 + mass of C from CO

  = 0.0481 kg/kg of flue gas

  Carbon in 1 kg of fuel = 0.82 kg

  Mass of dry flue gases formed per kg of fuel, m_g = = 17.048 kg

  Water formed by 0.05 kg of H_2 per kg of fuel =

$0.05 \times 9 \times 0.45$ kg

Mass of combustible matter per kg of fuel + air supplied per kg of fuel = mass of products

$$0.82 + 0.05 + 0.08 + m_a = 17.048 + 0.45$$

or m_a = = 16.55/kg fuel

-
- Theoretical air required for complete combustion of fuel

Excess air supplied = 16.55 – 10.9 = 5.65 kg of fuel

Percentage of excess air = × 100 = 51.83%

- Heat carried by flue gasses = Heat carried by dry flue gases + Heat carried by steam formed

$$= (1 \times 17.048 \times 380) + [2460 + 2.1 \times (380 - 15) \times 0.5183]$$
$$= 9335.5 \text{ kJ/kg of fuel burned}$$

- Carbon in fuel burned to CO =

The burning of carbon to CO instead of CO_2 causes a loss of (35000 – 10200)

$$= 24800 \text{ kJ/kg of C burned to CO}$$

Heat loss = 24800×0.1367 = 3389.3 kJ/kg of fuel

Example 1.6

The flue gas analysis by volume for burning of C_6H_{14} is as follows:

$$CO_2 = 8.5\%,\ CO = 7.8\%,\ \text{and}\ N_2 = 83.7\%$$

Calculate (a) the air-fuel ratio required for chemically correct combustion, and (b) the air-fuel (A:F) ratio as a percentage of the chemically correct mixture A:F ratio.

Solution

$$2C_6H_{14} + 19O_2 = 12CO_2 + 14H_2O$$
$$2 \times 86 + 19 \times 32 = 12 \times 44 + 14 \times 18$$

For complete combustion of 1 kg of fuel, oxygen required

∴ A:F ratio for complete combustion = 15.37: 1

The percentage of carbon in the fuel

= = 83.72%

Mass of air supplied per kg of fuel burned,

∴ A:F ratio of the mixture supplied

= 13.03:1

Example 1.7

The volumetric analysis of a sample of producer gas is as follows:

H_2 = 20%, CH_4 = 3%, CO = 22%, CO_2 = 8%, and N_2 = 47%.

- Find the volume of air required for complete combustion of 1 m^3 of the gas.
- If 50% excess air is supplied, find the percentage contraction in volume after the products of combustion have been cooled.

Solution

- Minimum volume of air required for complete combustion of 1 m^3 of fuel

$$= 1.29\ m^3/m^3 \text{ of gas}$$

- The constituents formed with 50% excess air are:

$CO_2 = 0.08 + CO_2$ formed from $CO + CO_2$ formed from CH_4

$$= 0.08 + 0.22 \times 1 + 0.03 \times 1 = 0.33\ m^3$$

$N_2 = 0.47 + N_2$ from the air supplied

$$= 0.47 + 1.29 \times 1.5 \times 0.79 = 2\ m^3$$

$O_2 = O_2$ from the excess air only

$$= (1.29 \times 0.5) \times 0.21 = 0.135\ m^3$$

Total dry flue gases formed $= CO_2 + N_2 + O_2 = 0.33 + 2.00 + 0.135 = 2.465\ m^3$

Total volume before combustion = volume of gas + volume of air supplied

$$= 1 + 1.29 \times 1.5 = 2.935\ m^3$$

Percentage contraction in volume after combustion and cooling

Example 1.8

A sample of coal has the following percentage composition by weights:

$$C = 82\%,\ H_2 = 10\%,\ O_2 = 6\%,\ N_2 = 1\%,\ \text{and}\ S = 1\%.$$

Calculate the minimum quantity of air required, for the complete combustion of 1 kg of coal. If 20% excess air is used in the combustion, what is the air-fuel ratio?

Solution

Minimum air per kg of fuel

With 20% excess air,

$$A{:}F = 12.765 \times 1.2 = 15.318\ \text{kg/kg of fuel}$$

Example 1.9

A fuel gas has composition by volume: H_2 = 50%, CO = 40%, CO_2 = 6% and N_2 = 4%. (a) Calculate the volume of air required for complete combustion of 1 m^3 of the gas, (b) If the dry exhaust gas from an engine using this fuel contains 9.2% by volume of CO_2 and no CO, calculate the air to gas ratio which has been used. Calculate the percentage of O_2 by volume in the dry exhaust.

Solution

- For complete combustion of fuel gas, the chemical equation is:

  where oxygen – fuel ratio by volume.

  Comparing coefficients of H_2, C and O_2 on both sides, we get

  $$b = 50, a = 40 + 6 = 46$$

  $x = 45$ m^3 of O2 per 100 m^3 of fuel gas.

A:F = = 2.14 m^3 of air/m^3 of gas.

•

where = new oxygen – fuel ratio by volume

C balance: $d = 40 + 6 = 46$

CO_2 balance: 0.092 =

O_2 balance: $+ 6 + y = 46 + e$

Solving for y, we get $y = 104$ and $e = 59$

Air to gas ratio by volume =

% volume of O_2 in dry gas = × 100 = 11.8%

Example 1.10

The composition of a hydrocarbon fuel by mass is: C = 85%, H_2 = 13%, and O_2 = 2%. Calculate (a) the chemically correct mass of air required for the complete combustion of 1 kg of fuel, (b) the

volumetric and gravimetric analysis of the products of combustion, and (c) dew point temperature if the total pressure is 1 bar and (d) If the products of combustion are cooled to 15°C, calculate the mass of water vapour condensed.

Solution

$$C + O_2 \rightarrow CO_2$$

Moles of O_2 required = = 0.103958 moles of O_2/kg of fuel

- Mass of air required = 0.103958 × × 29 = 14.356 kg of air/kg fuel
- The volumetric and gravimetric analysis of products of combustion:
- Partial pressure of H_2O, × p = 0.1234 × 1 = 0.1234 bar

  Dew point is the saturation temperature corresponding to 0.1234 bar. From steam tables,

  $$t_{dp} = 49.97°C.$$

- Temperature of products of combustion on cooling (15°C) is

less than t_{dp}. Hence, some water vapour will condense. The vapour pressure will then be equal to the saturation vapour pressure corresponding to 15°C.

From steam tables, $p_s = 0.017051$ bar and $v_g = 77.926$ m^3/kg.

The volume occupied by the water vapour is equal to the volume occupied by CO_2 and N_2 at their partial pressure of (1 – 0.017051) = 0.982949 bar.

Volume, = 11.25 m^3

Mass of vapour in combustion products = = 0.14437 kg

Mass of condensed vapour = 1.17 – 0.14437 = 1.0256 kg

Example 1.11

Octane, C_8H_{18}, is burned with dry air. The molar analysis of the dry products is CO_2 = 10%; CO = 0.4%; O_2 = 3%, and N_2 = 86.6%. Determine the air fuel ratio on a

mass basis and the dew point temperature of the products if the products are at 1 atm pressure. Dry air contains 3.76 moles of nitrogen per mole of oxygen.

[IES, 1994]

Solution

Mass of C per kg of dry flue gas = Mass of C from CO_2 + Mass of C from CO

Percentage of C in fuel = × 100 = 84.21%

Mass of dry flue gases formed per kg of fuel = = 20.05 kg

Mass of air supplied per kg of fuel =

Air/Fuel ratio = 21.25:1.

The dew point temperature of products is the saturation temperature at the partial pressure of water vapour.

$$C_8H_{18} + 12.5O_2 + 12.5\,(3.76)\,N_2 \rightarrow 8CO_2 + 9H_2O + 47N_2$$

12.5 moles of O_2 are required to burn 1 mole C_8H_{18} and 8 moles of H_2O are produced.

Stoichiometric mole fraction of water vapour (xH_2O) formed,

$$pH_2O = xH_2O \text{ atm} = 0.1636 \text{ bar}$$

The saturation temperature of water vapour corresponding to their partial

pressure of 0.1636 bar is, $t_s = 54°C$.

Example 1.12

An oil fired boiler uses a fuel which contains 84% carbon and 16% hydrogen by weight. The volumetric analysis of the dry flue gases was found to be CO_2 = 9%, CO = 2%, O_2 = 5%, and N_2 = 84%. Assuming complete combustion of the hydrogen in the fuel, determine the quantity of carbon remaining unburnt in each kg of the fuel fired in the furnace. Determine the actual air-fuel ratio and the excess air factor.

[IES, 1983]

Solution

$$C + O_2 = CO_2$$
$$12 + 32 = 44$$

0.84 kg of C requires × 0.84 = 2.24 kg of O_2

$$2H_2 + O_2 = 2H_2O$$
$$4 + 32 = 36$$

0.16 kg of H_2 requires × 0.16 = 1.28 kg of O_2

Total O_2 required for complete combustion of 1 kg of fuel = 2.24 + 1.28 = 3.52 kg

Amount of air required for complete combustion of 1 kg of fuel

Air/fuel ratio = 15.3:1

Mass analysis of dry flue gases:

Mass of C per kg of dry flue gas

Mass of C per kg of fuel = 0.84 kg

Mass of dry flue gases per kg of fuel = = 18.876 kg

Mass of N_2 per kg of fuel = 18.876 × 0.7935 = 14.978 kg

Mass of air required per kg of fuel = = 19.45 kg

Excess air supplied = 19.45 – 15.3 = 4.15 kg

Excess air factor = = 0.2714 or

27.14%

Example 1.13

The composition of coal burnt during a boiler trial was as follows:

$C = 62\%$, $H_2 = 3.6\%$, $O_2 = 7.4\%$, and ash = 27%.

The volumetric analysis of the fuel gas showed the following:

$CO_2 = 10.25\%$, $CO = 0.80\%$, $O_2 = 8.54\%$, and $N_2 = 80.41\%$.

Determine

1. the air-fuel ratio
2. the percentage of excess air, and
3. the mass of CO_2, CO, O_2, and N_2 in the flue gases per kg of coal burnt.

[IES, 1986]

Solution

Conversion of volumetric analysis to mass analysis.

- Mass of carbon per kg of dry flue gas
  = mass of C in CO_2 + mass of C in CO

  Mass of dry flue gases formed per kg of fuel

  Mass of steam formed per kg of fuel = 0.036×9 = 0.324 kg

  Mass of combustible matter + air supplied per kg of fuel = mass of products C + H_2 + O_2 + air supplied = mass of products

  $0.62 + 0.036 + 0.074 + m_a = 14.03 + 0.324$

  or $m_a = 13.624$ kg/kg of fuel

  Air/fuel ratio = 13.624:1.

- Theoretical air required
  = 8.12 kg/kg of fuel

  Excess air supplied = 13.624 – 8.12 = 5.504 kg/kg of fuel

  Percentage of excess air supplied = = 67.8%

- Flue gas analysis

Example 1.14

The mass analysis of a hydrocarbon fuel is as follows:

$C = 84\%$, $H_2 = 5\%$ and the balance is incombustible material. Find (a) the mass of air required per kg of fuel for complete combustion, (b) the analysis of the wet exhaust gases, by mass and volume, if 20 kg fuel is supplied, (c) the partial pressure of the steam formed in the exhaust gases if the total pressure of the exhaust gases is 103 kPa, (d) the heat carried away by dry exhaust gases formed per kg of fuel if the temperature of exhaust gas is 375°C and the ambient temperature is 24°C. Take c_p for dry gases is 1.005

kJ/kgK.

[IES, 1988]

Solution

- Oxygen required for complete combustion of 1 kg of fuel

  Air required per kg of fuel for complete combustion

- The products of combustion per kg of fuel are:

$$N_2 = 14.96 \times 0.77 = 11.52 \text{ kg}$$

  Wet products

- Partial pressure of steam formed in the exhaust gases
- Mass of dry exhaust gases formed per kg of fuel

$$= (\text{mass of } CO_2 + \text{mass of } N_2) \text{ per kg of fuel}$$
$$= 3.08 + 11.52 = 14.60 \text{ kg}$$

  Heat carried away by dry exhaust gases per kg of fuel burned

$$= 14.60 \times 1.005 \times (375 - 24) = 5150.223 \text{ kJ}$$

Example 1.15

A sample of gaseous fuel has the

following composition by volume:

H_2 = 28%, CO = 10%, CH_4 = 2%, CO_2 = 16%, O_2 = 2%, N_2 = 42%.

If this fuel is burned with 50% excess air, determine (a) the volume of air per cubic metre of gas at the same temperature and pressure, (b) the volumetric analysis of dry products, (c) the mass of products per kg of fuel, (d) the mass of dry products per kg of fuel, and (e) the mass of air supplied per kg of fuel.

Solution

- Minimum volume of air required for complete combustion of 1 m^3 of fuel,

  = 1.095 m^3/m^3 of gaseous fuel

  Air required with 50% excess air = 1.095 × 1.5 = 1.643 m^3/m^3 of gas

- The constituents formed with 50% excess air are:

$CO_2 = 0.16 + CO_2$ formed from $CO + CO_2$ formed from CH_4

$= 0.16 + 0.10 \times 1 + 0.02 \times 1 = 0.28\ m^3$

$N_2 = 0.42 + N_2$ from air supplied $= 0.42 + 1.643 \times 0.79 = 1.718\ m^3$

$O_2 = 0.02 + O_2$ from excess air supplied $= 0.02 + 1.095 \times 0.5 \times 0.21 = 0.135\ m^3$

Total dry flue gases formed $= CO_2 + N_2 + O_2 = 0.28 + 1.718 + 0.135 = 2.133\ m^3$

-
- Mass of air supplied per kg of fuel =
- Mass of air supplied =
  = 0.6318 kg/kg of fuel

With 50% excess air, mass of air supplied = $0.6318 \times 1.5 = 0.9478$ kg/kg of fuel

Example 1.16

A hydrocarbon fuel when burned with air gave the following Orsat

analysis:

$CO_2 = 11.94\%$, $O_2 = 2.26\%$, $CO = 0.41\%$, $N_2 = 85.39\%$

Determine (a) the air-fuel ratio on mass basis, (b) the percent of carbon and hydrogen in the fuel on mass basis, and (c) the percent of theoretical air supplied. Assume that air contains 21% oxygen by volume.

[IES, 1998]

Solution

Let 100*x* moles of air be used with fuel per 100 moles of dry exhaust products.

$(aC + bH_2)_{fuel} + (21 \times O_2 + 79 \times$

$N_2)_{air}$ = 11.94 CO_2 + 2.26 O_2 + 0.41 CO + 85.39 N_2 + $d H_2O$

By molar balance, we have

Carbon: $a = 11.94 + 0.41 = 12.35$

Hydrogen: $b = d$

Oxygen: $21x = 11.94 + 2.26 +$

$$= 14.405 + 0.5d$$

Nitrogen: $79x = 85.39$ or $x = 1.08$

From the above equation, we get

$21 \times 1.081 = 14.405 + 0.5d$

$d = b = 16.592$

- Air-fuel ratio = = 17.28
- % Carbon in fuel = = 81.7%

% Hydrogen in fuel = = 18.3%

- % Theoretical air supplied + (21 × O_2 + 79 × N_2) = 94.5%

Example 1.17

The chemical formula for alcohol is C_2H_6O. Calculate the stoichiometric air/fuel ratio by mass and the percentage composition of the products of combustion per kg of C_2H_6O.

[GATE, 1998]

Solution

The chemical equation for complete combustion of given fuel can be written as follows:

$$C_2H_6O + 3O_2 \rightarrow 2CO_2 + 3H_2O$$

$$(1 \times 46) + (3 \times 32) = (2 \times 44) + (3 \times 18)$$

For complete combustion of 1 kg of C_2H_6O, oxygen required

$$= 2.087 \times = 9.074 \text{ kg of air}$$

∴ A:F ratio for complete combustion = 9.074: 1

Also 46 kg of fuel produces products of combustion = 88 + 54 = 142 kg

∴ 1 kg of fuel produces = 3.087 kg of products of combustion

(i.e., CO_2 and H_2O)

Hence, CO_2 produced by fuel = × 100 = 1.913 or 191.3%

H_2O produced by fuel = × 100 = 1.174 or 117.4%

Example 1.18

The products of combustion of an unknown hydrocarbon C_xH_y have the following composition as measured by an Orsat apparatus.

CO_2 = 8.0%, CO = 0.9%, O_2 = 8.8%, and N_2 = 82.3%.

Determine (a) the composition of the fuel (b) the air-fuel ratio; and (c) the percentage of excess air used.

Solution

- Let *a* moles of oxygen by supplied per mole of fuel. The chemical reaction can be written as follows:

$$C_xH_y + aO_2 + 3.76\,aN_2 \rightarrow 8CO + 0.9CO_2 + 8.8O_2$$

$+ 82.3H_2 + H_2O$

Nitrogen Balance gives $3.76a = 82.3$

or $a = 21.89$

Oxygen Balance gives: $2a = 16 + 0.9 + 17.6 +$

or $y = 18.5$

Carbon balance gives

$$x = 8 + 0.9 = 8.9$$

Therefore, the chemical formula of the fuel is $C_{8.9}H_{18.5}$.

The composition of the fuel is

% Carbon = × 100 = 85.23%

% Hydrogen = 14.77%

- Air fuel ratio =
- % Excess air used × 100 = 67.22%

Example 1.19

The volumetric analysis of a gas is

14% CO_2, 1% CO, 5% O_2, and 80% N_2. Calculate the fuel gas composition by mass.

Solution

Given: CO_2 = 14% = 0.14 m^3; CO = 1% = 0.01 m^3; O_2 = 5% = 0.05 m^3; N_2 = 80% = 0.8 m^3

The volumetric analysis may be converted into mass analysis by completing the table as follows:

The fuel gas composition, by mass is given in the last column, i.e., CO_2 = 20.2%, CO = 0.9%, O_2 = 5.3%, and N_2 = 73.6%.

Example 1.20

A blast furnace gas has the following volumetric composition:

$CO_2 = 11\%$, $CO = 27\%$, $H_2 = 2\%$, and $N_2 = 60\%$.

Find the theoretical volume of air required for the complete combustion of 1 m^3 of the gas. Find the percentage composition of dry flue gases by volume. Assume that air contains 21% of O_2, and 79% of N_2 by volume.

Solution

Given: $CO_2 = 11\% = 0.11\ m^3$, $CO = 27\% = 0.27\ m^3$, $H_2 = 2\% = 0.02\%\ m^3$, $N_2 = 60\% = 0.6\ m^3$

We know that theoretical volume of air required

($\therefore$ CH_4, C_2H and O_2 are equal to zero)

We know that 1 m^3 CO produces 1 m^3 of CO_2, therefore

Volume of CO_2 obtained from the given 0.27 m^3 of CO = 0.27 m^3

and volume of CO_2 already present in the fuel = 0.11 m^3 (Given)

$\therefore$ Total volume of CO_2 in the flue gas = 0.11 + 0.27 = 0.38 m^3

Now volume of N_2 from the theoretical air supplied = × 0.69 =

0.545 m^3

and volume of N_2 already present in the fuel = 0.6 m^3

∴ Total volume of N_2 in the flue gas = 0.6 + 0.545 = 1.145 m^3

and volume of the dry flue gas = Total volume of CO_2 + Total volume of N_2

$$= 0.38 + 1.145 = 1.525 \text{ m}^3$$

∴ Percentage of CO_2 in the dry flue gas = × 100 = 24.92%

and percentage of N_2 in the dry flue gas =× 100 = 75.08%

Example 1.21

The percentage composition by mass of a sample of coal as found by analysis is given as: C = 90%, H_2 = 3.3%, O_2 = 3.0%, N_2 = 0.8%, S = 0.9%, and ash 2.0%.

Calculate the minimum mass of air required for the complete combustion of 1 kg of this fuel. If 50% excess air is supplied, find the total mass of dry flue gases per kg of fuel and the percentage composition of the dry flue gases by volume.

Solution

Given: C = 90% = 0.9 kg; H_2 = 3.3% = 0.033 kg; O_2 = 3% = 0.03

kg; N = 0.8% = 0.008 kg; S = 0.9% = 0.009 kg; Ash = 2% = 0.02 kg; Excess air supplied = 50%

We know that minimum mass of air required for complete combustion of 1 kg of fuel.

Total mass of dry flue gases per kg of fuel

Since 50% excess air is supplied, therefore actual amount of air supplied per kg of coal = 11.5 × 1.5 = 17.25 kg

∴ Excess air supplied = 17.25 – 11.5 = 5.75 kg

The products of combustion are

represented by the following chemical equations:

$$C + O_2 = CO_2$$
$$2H_2 + O_2 = 2H_2O$$
$$S + O_2 = SO_2$$

In addition to carbon dioxide, water, and sulphur dioxide, the excess oxygen and nitrogen will be available in the products of combustion. It may be noted that H_2O (water vapour) is a wet gas, therefore, the dry flue gases are only carbon dioxide, sulphur dioxide, excess oxygen, and nitrogen. Let us now find the mass of each of these flue gases per kg of fuel.

We know that 1 kg of carbon produces 11/3 kg of carbon dioxide and 1 kg of sulphur produces 2 kg

of sulphur dioxide.

∴ Mass of CO_2 contained in 0.9 kg of carbon per kg of fuel = × 0.9 = 3.3 kg(i)

and mass of SO_2 in 0.009 kg of sulphur per kg of fuel = 2 × 0.009 = 0.018 kg(ii)

We also know that the mass of excess O_2 per kg of fuel = × Excess air supplied

and mass of nitrogen in the products of combustion per kg of fuel

∴ Total mass of dry flue gases per kg of fuel = 3.3 + 0.018 + 1.323 +

13.283 kg

$$= 17.924 \text{ kg}$$

First, let us find out the percentage composition of dry flue gases from the aforementioned data by mass. We know the following composition:

Now, let us convert this mass analysis of dry flue gases into volumetric analysis as follows:

The percentage composition of dry flue gases (by volume) is given in last column, that is, CO_2 = 12.68%,

SO_2 = 0.06%; Excess O_2 = 7.01%; and N_2 = 80.25%.

C = 85%, H_2 = 12.5%, O_2 = 2% and the residue 0.5%.

Example 1.22

A liquid fuel C_7H_{16} is burned with 10% more air than the stoichiometric air. Assuming complete combustion, calculate (a) the mass of air supplied per kg of fuel and (b) the volumetric analysis of the dry products of combustion. Assume air contains 23% O2 by mass.

[IES, 2009]

Solution

- The chemical equation of combustion for the given fuel is:

  $C_7H_{16} + 11O_2 = 7CO_2 + 8H_2O$

  100 kg + 11 × 32 kg = 308 kg + 144 kg

  1 kg of fuel requires = kg of O_2

  = 15.3 kg of air/kg of fuel

  When 10% more air than the stoichiometric air is burned, then actual mass of air supplied = 1.1 × 15.3 = 16.83 kg of air/kg of fuel

  Mass of CO_2 formed = = 3.08 kg/kg of fuel

  O_2 comes in exhaust from complete air supplied,

  N_2 = 16.83 × = 12.96 kg/kg of fuel

  O_2 comes in exhaust from only excess air supplied,

  O_2 = 15.3 × 0.1 × = 0.352 kg

- The mass is converted into volume as listed below in the table:

Example 1.23

A four-cylinder engine of an automobile is converted to run on propane (C_3H_8) fuel. A dry analysis of engine exhaust gives volumetric percentage of CO, CO_2 and O_2, respectively at 9.79%, 4.90% and 2.45%. Write the resulting chemical reaction and find the equivalence ratio.

[IES, 2010]

Solution

Given: Volumetric analysis: CO = 9.7%, CO_2 = 4.90%, O_2 = 2.45%

The volumetric analysis of exhaust analysis can be converted to mass analysis as follow:

Percentage of C in C_3H_8 = = 0.8182 or 81.82%

Percentage of H_2 in C_3H_8 = 100 – 81.82 = 18.18%

Mass of air supplied per kg of fuel,

Oxygen required for complete combustion of 1 kg of fuel

Minimum air required for complete combustion of fuel

Example 1.24

Coal, having the following composition, is burnt in a furnace:

Carbon = 86.1%, hydrogen = 3.9%, oxygen = 1.4%, rest ash.

The volumetric analysis of dry products was as follows:

CO_2 = 12.7%, CO = 1.4%, O_2 = 4.1%, N_2 = 81.8%. Determine the percentage of excess air. Determine the loss due to incomplete combustion, if CO burning to CO_2 will release 24000 kJ/kg of carbon in CO.

[IAS, 1998]

Solution

Given: Composition of coal: C = 86.1%, H_2 = 3.9%, O_2 = 1.4%, Rest = 8.6% ash

Volumetric analysis of dry products:

CO_2 = 12.7%, CO = 1.4%, O_2 = 4.1%, N_2 = 81.8%.
$2CO + O_2 \rightarrow 2CO2$ + 24,000 kJ/kg of C in CO

Mass of air supplied per kg of fuel,

Theoretical air required for complete combustion of fuel

= 11.278 kg/kg of fuel

Excess air supplied = 15.927 – 11.278 = 4.649 kg/kg of fuel.

Carbon in fuel burned to = 0.10448 kg

Loss due to incomplete combustion = 0.10448 × 24000

= 2507.5 kJ/kg of fuel.

Example 1.25

The mass analysis of the petrol used in an engine was 85% C and 15% H_2. The dry exhaust gas analysis showed that the percentage by volume of carbon dioxide was six times that of oxygen and that no carbon-monoxide was present. Calculate: (a) the air-fuel ratio by mass and (b) the percentage excess air supplied.

Assume air contains 23.2% O_2 by mass or 20.9% O_2 by volume.

[IAS, 1999]

Solution

Given: C = 0.85, H_2 = 0.15% by

mass

- Maximum mass of air required for complete combustion,

  = 14.94 kg

The combustion equation for the given fuel can be written as:

where b, c, d and e are the products formed and 'a' is the amount of O_2 supplied per kg of fuel.

Balancing the mass of carbon from both sides, we have

$b \times = 0.85$ as 44 kg of CO_2 contains 12 kg of carbon.

$b = 0.85 \times = 3.117$ kg

Balancing the mass of hydrogen from both sides, we have

$e \times = 0.15$ as 18 kg of H_2O contains 2 kg of hydrogen.

$e = 1.35$ kg

Balancing the mass of O_2 from both sides, we have

$a = b \times + c + e \times$ as 44 kg of CO_2 contains 32 kg

of O_2 and 18 kg of water contains 16 kg of O_2.

$a = 0.7273\, b + c + 0.889\, e$

Now balancing the mass of N_2 from both sides, we have

$a \times = d$

$a = 0.302\, d$

Thus, we have

$a = 0.7273 \times 3.117 + c + 0.889 \times 1.35$

$= 2.267 + c + 1.2$

$= 3.467 + c$

Volumetric analysis of exhaust gases:

As per the given condition, percentage by volume of CO_2 = 6 × percentage by volume of O_2

or $0.0708 = 6 \times$

$c = 0.3776$ kg of O_2 in the exhaust gases when one kg of fuel is burnt.

We have $a = 3.467 + 0.3776 = 3.8446$ kg of O_2 is supplied per kg of fuel.

Amount of actual air supplied per kg of fuel,
$(m_a)_{act} = 3.8446 \times = 16.57$ kg

$\therefore$ A:F ratio = 16.57:1

- % Excess air supplied = = 0.1091 or 10.91%

Example 1.26

A four cylinder engine of a truck has been converted to run on propane fuel. A dry analysis of the engine exhaust gives the following volumetric percentages:

$CO_2 = 4.90$; $CO = 9.79$ and $O_2 = 2.45$.

Calculate the equivalence ratio at which the engine is operating.

[IAS, 2001]

Solution

Given: Volumetric analysis of engine exhaust: CO_2 = 4.90%, CO = 9.79%, O_2 = 2.45%

Mass of air supplied, m_a =

Propane: $C_5H_{12} + 8O_2 = 5CO_2 + 6H_2O$

Minimum amount of air required in kg per kg of fuel for complete combustion

Actual A:F ratio = 15.46:1

Stoichiometric amount of air required per kg of fuel burned,

Summary for Quick Revision

1. Fuels may be classified as solid, liquid, and gaseous fuels.
2. The calorific value of a solid or liquid fuel is the heat evolved by the complete combustion of unit mass of the fuel.
3. The calorific value of a gaseous fuel is the heat evolved by the complete combustion of one cubic metre of gas at STP.
4. Higher calorific value (HCV) of a fuel is obtained when the water vapour produced during the process of combustion due to hydrogen present in the fuel is allowed to condense evolving their latent heat.
5. Lower calorific value (LCV) of fuel is obtained when the latent heat of water vapour goes waste.
6. LCV = HCV – m_w LCV_{fg}

   where m_w = mass of water vapour formed per kg of fuel burned.

   h_{fg} = latent heat of vapourisation at the partial pressure of water vapour in the products of combustion

7. HCV = where C, H, O, S = percentage of carbon, hydrogen, oxygen, and sulphur in the fuel.
8. LCV = HCV – × H × H.V. = HCV
9. Combustion of hydrocarbon fuel:

   $x\text{C}_n\ \text{H}_{2n} +_2 + y\text{O}_2$ O $a\ \text{CO}_2 + b\text{H}_2\text{O}$

   where $a = xn$, $b = x(n + 1)$, $y = (3n + 1)$

10. Minimum air required for complete combustion of solid/liquid fuels
11. Mass of air supplied per kg of fuel,

    where C = mass fraction of carbon in fuel

C_1, C_2 = fraction of CO and CO_2 by volume in dry flue gases.

12. Excess air supplied = kg
13. Percentage of C burning to CO_2 =

Percentage of C burning to CO =

14. The Orsat apparatus is used for flue gas analysis. The percentage of SO_2 cannot be measured by this apparatus. NaOH or KOH is used to absorb CO_2, alkaline solution of pyrogallic acid absorbs O_2, and cuprous chloride absorbs CO_1. N_2 is obtained by difference.
15. The bomb calorimeter is used to measure different HCVs of solid and liquid fuels.

HCV =

where m_f, m_{fw}, m_w, m_c = mass of fuel, fuse wire, water in calorimeter, and water equivalent of calorimeter, respectively.

$\Delta\theta$ = true temperature rise

C.V. = calorific value of fuse wire.

16. The Boys gas calorimeter is used to determine the HCV of gaseous fuels.

HCV = kJ/m^3 at NTP

where $(\Delta\theta)_w$ = rise in temperature of water circulated,

m_w = mass of water circulated, kg

V_g = volume of gas at NTP

Multiple-choice Questions

1. The percentage of oxygen contained in air on volume basis is
    1. 21
    2. 23
    3. 79
    4. 77
2. The percentage of oxygen contained in air on mass basis is
    1. 21
    2. 23
    3. 79
    4. 77
3. The mass of oxygen required to convert 1 kg of carbon to CO_2 is will be
    1. kg
    2. kg
    3. kg
    4. kg
4. A gas is formed by mixing equal masses of O_2 and N_2 gas. The ratio of N_2 and O_2 by volume will be
    1.
    2.
    3.
    4.
5. CO produced by 1 kg of C is
    1. kg
    2. kg
    3. kg
    4. kg
6. The ratio of air required for complete combustion of C to CO_2 as compared to that for CO is
    1. 1.5:1
    2. 2:1
    3. 3:1
    4. 4:1
7. Which law is associated with combustion of gaseous fuels?
    1. Avogadro's hypothesis
    2. Dalton's law
    3. Charles's law

4. Boyle's law

8. The Orsat apparatus is used for
   1. gravimetric analysis of flue gases
   2. volumetric analysis of flue gases
   3. determining mass flow rate of flue gases
   4. measuring smoke density of flue gases
9. Alkaline pyrogallol is used in Orsat apparatus for absorption of
   1. CO_2
   2. CO
   3. O_2
   4. N_2
10. The proximate analysis of fuel is determination of percentage of
    1. C, H_2, N_2, S, and moisture
    2. fixed carbon, ash, volatile matter, and moisture
    3. higher calorific value
    4. lower calorific value
11. Bomb calorimeter is used to determine
    1. HCV at constant volume
    2. HCV at constant pressure
    3. LCV at constant volume
    4. LCV at constant pressure
12. The ultimate analysis of fuel is determination of
    1. C, H_2, N_2, S, and moisture
    2. fixed carbon ash, volatile matter, and moisture
    3. higher calorific value
    4. lower calorific value
13. Pour point of a fuel oil is the
    1. lowest temperature at which oil will flow of its own
    2. storage temperature
    3. temperature of transportation of oil
    4. temperature of pumping of oil through burners
14. Orsat apparatus is used to determine in flue gases the percentage of
    1. CO_2, CO, and O_2
    2. CO_2, CO, O_2, and N_2
    3. CO_2
    4. CO
15. In the Orsat apparatus, CO_2, O_2 and CO are absorbed in first three pairs of bulbs by filling them with
    1. sodium hydroxide solution, pyrogallol, and cuprous chloride solutions, respectively
    2. pyrogallol, sodium hydroxide, and cuprous chloride,

respectively
3. cuprous chloride, sodium hydroxide, and pyrogallol, respectively
4. sodium hydroxide, cuprous chloride, and pyrogallol, respectively

16. Orsat apparatus is employed to determine
    1. ultimate analysis of fuel
    2. gravimetric analysis of fuel
    3. volumetric analysis of dry products of combustion
    4. gravimetric analysis of dry products of combustion
17. In the Orsat apparatus,
    1. CO_2 is absorbed in cuprous chloride
    2. CO is absorbed in caustic potash solution
    3. O_2 is absorbed in pyrogallic acid
    4. N_2 is absorbed in hot nickel-chrome compound
18. Items given in List I and II pertain to gas analysis. Match List I with II and select the correct answer.

Codes:

	A	B	C	D
1.	2	3	1	4
2.	1	3	2	4
3.	1	5	4	2
4.	2	5	4	1

19. If methane undergoes combustion with the stoichiometric quantity of air, the air-fuel ratio on molar basis would be
    1. 15.22: 1
    2. 12.30: 1
    3. 14.56: 1
    4. 9.52: 1
20. The presence of nitrogen in the products of combustion ensures that
    1. complete combustion of fuel takes place
    2. incomplete combustion of fuel occurs
    3. dry products of combustion are analysed
    4. air is used for combustion
21. In the combustion process, the effect of dissociation is to

1. reduce the flame temperature
2. separate the products of combustion
3. reduce the proportion of carbon monoxide in gases
4. reduce the use of excess air

22. Volumetric analysis of sample dry products of combustion gave the following results:

CO_2 = 10% CO = 1% O_2 = 8% N_2 = 81%

Their proportion by weight will be

1. 440: 28: 256: 2268
2. 22: 14: 256: 1134
3. 440: 14: 28: 2268
4. 22: 28: 14: 1134

23. The minimum weight of air required per kg of fuel for complete combustion of a fuel is given by
    1. 11.6C + 34.8 + 4.35 S
    2. 11.6C + 34.8 + 4.35 S
    3. 11.6C + 34.8 + 4.35 S
    4. 11.6C + 34.8 + 4.35 S
24. The mass of air required for complete combustion of unit mass of fuel can always be calculated from the formula, where C, H, O and S are in percentage. Which of the following is the correct option?
    1. 0.1152C + 0.3456H
    2. 0.1152C + 03456 (H – 0.125O)
    3. 0.1152C + 0.3456 (H – 1.25O) + 0.0432S
    4. 0.1152C + 0.3456 (H + 0.125) + 0.0432S
25. List I gives the different terms related to combustion while List II gives the outcome of the events that follows. Match List I with List II and select the correct answer.

Codes:

	A	B	C	D
1.	3	4	1	2

2. 4 3 1 2
3. 3 4 2 1
4. 4 3 2 1

26. The calorific value determined by the bomb calorimeter is
    1. lower calorific value at constant pressure
    2. higher calorific value at constant pressure
    3. lower calorific value at constant volume
    4. higher calorific value at constant volume
27. Incomplete combustion is indicated by
    1. high percentage of carbon monoxide in the exhaust gases
    2. high percentage of carbon dioxide in the exhaust gases
    3. high temperature of exhaust gases
    4. low temperature of exhaust gases
28. In order to burn 1 kg of CH_4 completely, the minimum number of kilograms of oxygen needed is (take atomic weights of H, C, and O as 1, 12, and 16 respectively).
    1. 3
    2. 4
    3. 5
    4. 6
29. Consider the following statements:
    1. For the combustion of pulverised coal, 5%–10% excess air is required.
    2. Air contains 21% oxygen by weight.
    3. The flue gases from a coal-fired furnace contain around 70% nitrogen by volume.
    4. In the combustion of liquid fuels, the number of moles of the products are invariably greater than the number of moles of the reactants.

    Of these statements:

    1. I, II, and IV are correct
    2. I, III, and IV are correct
    3. II, III, and IV are correct
    4. I and III are correct
30. Which one of the following gaseous fuels does not have different higher and lower calorific values?
    1. Methane
    2. Ethane
    3. Carbon monoxide

4. Hydrogen

Review Questions

1. What is the proximate analysis of coal?
2. What do you understand by ultimate analysis of coal?
3. What is a bomb calorimeter?
4. How does the LCV of a fuel differ from its HCV?
5. What is a pour point of a fuel?
6. What is the flash point?
7. What is LPG?
8. What is combustion?
9. Why is excess air always required to be supplied for combustion of fuel?

Exercises

1.1 Calculate the HCV and LCV of a fuel having the following composition C = 80%, H_2 = 10%, O_2 = 3%, S = 2%, N_2 = 20% and rest in combustible matter.

[**Ans.** 41946.95 kJ/kg, 39732.95 kJ/kg]

1.2 The following data refer for the determination of calorific value of a gaseous fuel:

Barometer pressure = 740 mm Hg

Volume of fuel used = 15 litres

Pressure of gas = 40 mm of water above atmospheric

Temperature of gas = 20°C

Volume of cooling water circulated = 12.5 litres

Rise in temperature of cooling water = 5°C

Volume of condensate collected = 0.01 litres

Latent heat of steam = 2460 kJ/kg

Specific gravity of Hg = 13.6

Calculate HCV and LCV of the fuel.

[**Ans.** 19213.3 kJ/kg, 16512 kJ/kg]

1.3 The following data refer to determination of calorific value of coal by using bomb calorimeter:

Mass of coal sample = 1.01 gram

Mass of water = 2500 gram

Water equivalent of calorimeter = 744 gram

Temperature rise of water = 2.59°C

Temperature correction for cooling = 0.01°C

Calculate HCV of coal.

[**Ans.** 35073.7 kJ/kg]

1.4 The gravimetric analysis of a sample coal is given as follows: C = 80%, H_2 = 12%, and ash = 8%. Calculate the stoichiometric A:F ratio and the analysis of products by volume.

[**Ans.** 13.3: 1, CO_2, = 13.6%, H_2 = 12.2%, N_2= 74.2%]

1.5 A sample of coal gave the following ultimate analysis by mass: C = 81.9%, H_2 = 4.9%, O_2 = 6%, N_2 = 2.3%, ash = 4.9%. Determine (a) the stoichiometric A:F ratio, and (b) the analysis by volume of the wet dry products of combustion when air supplied is 25% in excess of that required for complete combustion.

[**Ans.** 10.8: 1; CO_2 = 14.16%, H_2O = 5.07%, CO = 4.04%, N_2 = 76.73%; CO_2 = 14.9%, N_2, = 85.1%]

1.6 The dry exhaust analysis of a petrol

engine gave CO = 1.5% and O_2 was negligible if the fuel had an ultimate analysis by weight of C = 84% and H_2 = 16%, calculate the air–fuel ratio supplied.

[**Ans.** 14.75°C]

1.7 Methane is burnt with 50% excess air. The products of combustion exist at 1 bar, 250°C. Calculate the dew-point temperature of water vapours in the combustion products.

[**Ans.** 51°C]

1.8 The composition of a sample of coal was found to be: C = 89.1%, H_2 = 5%, O_2 = 4.2%, N_2 = 1.5%, and remainder ash. Calculate the stoichiometric air-fuel ratio by mass if 30% excess air is supplied, calculate the percentage

composition of dry gases by volume.

[**Ans.** 11.9: 1; CO_2 = 14.1%, O_2 = 4.9%, N_2 = 81%]

1.9 During a boiler test, it was found that flue gases were leaving the boiler at a temperature of 300°C and inlet air temperature was 20°C. The volumetric composition of the flue gases was: CO_2 = 11.6%, O_2 = 6.8, CO = 0.5%. The coal supplied to the furnace has 80% of carbon by mass. Calculate the heat carried away by the exhaust gases per kg of coal burned. Assume the average specific heat of flue gases to be 1 kJ/kg.

[**Ans.** 4647.3 kJ/kg of coal burned]

1.10 A sample of coal has the following composition by weight: C = 77.7%, H_2 = 6.8%, N_2 = 1.2%, S = 2.2%, O_2 = 8.8%, and incombustible matter = 3.3%.

Determine the minimum quantity of air required for complete combustion of 1 kg of coal and the constituent products of combustion in kg per kg of coal.

[**Ans.** 11.086 kg/kg of coal; CO_2 = 2.849, H_2O = 0.612, SO_2 = 0.044, N_2 = 0.012 kg/kg of coal]

1.11 The analysis of a coal contains; C = 82%, H_2 = 6%, O_2 = 4%, H_2O = 2%, and ash = 6%. Determine the theoretical minimum air required for complete combustion of 1 kg of coal. If the actual air supplied is 18 kg per kg of coal, the hydrogen is completely burned and 80% of C is burned to CO_2, and the remainder to CO, determine the volumetric analysis of the dry products of combustion.

[**Ans.** 11.412 kg/kg of coal; CO_2 = 8.84%, CO = 2.22%, O_2 = 8.74%, N_2 = 80.2%]

1.12 A coal has the following analysis: C = 81%, H_2 = 4.5%, O_2 = 8%, and remainder incombustible matter. The Orsat analysis of the dry flue gases was: CO_2 = 8.3%, CO = 1.4%, O_2 = 10%, and N_2 = 80.3% (by difference). Calculate the mass of air supplied per kg of coal and the percentage of excess air.

[**Ans.** 20.15 kg/kg of coal, 89.8%]

1.13 A coal of calorific value 29,725 kJ/kg has a composition by weight of C = 80%, H_2 = 5%, O_2 = 8%, S = 2%, N_2 = 2%, and remainder is ash. It is burnt in a furnace with 50% excess air. The flue gases enter the chimney at 325°C and the atmosphere temperature is 15°C. Calculate the proportion of heat carried away by the flue gases c_p for air = 1.005

kJ/kg.K and for dry products of combustion 1.046 kJ/kgK. Heat carried away by moisture in flue gases = 29.0 kJ per kg of moisture.

[**Ans.** 25.4%]

1.14 An unknown hydrocarbon fuel has the following Orsat analysis: CO_2 = 12.5%, CO = 0.3%, O_2 = 31%, N_2 = 84.1%. Determine the air-fuel ratio, fuel composition on mass basis, stoichiometric air-flue ratio, and percentage of excess air.

[**Ans.** 34.44: 1; C = 85.33%, H_2 = 14.67%; 15:1; 129.6%]

1.15 The percentage analysis of a gaseous fuel by volume is given as follows: CO_2 = 8%, CO = 22%, O_2 = 4%, H_2 = 30%, and N_2 = 36%. Determine the (a) the minimum volume

of air required for complete combustion of 1 m^3 of gas, (b) the percentage composition by volume of the dry products of combustion, and (c) if 1.4 m^3 of air is supplied per m^3 of gas, what will be the percentage by volume of CO_2 in the dry products of combustion?

[**Ans.** 1.05 m^3; CO_2 = 20.14%, N_2 = 79.86%; 16.3%]

1.16 During the trial on a boiler, the dry flue gas analysis by volume was obtained as: CO_2 = 13%, CO = 0.3%, O_2 = 6%, N_2 = 80.7%. The coal analysis by weight was reported as: C = 62.4%, H_2 = 4.2%, O_2 = 4.5%, moisture = 15%, and ash 13%. Determine (a) the theoretical air required to burn 1 kg of coal, (b) the mass of air actually supplied per kg of coal, and (c) the

amount of excess air supplied per kg coal burnt.

[**Ans.** 8.5 kg/kg of fuel, 11.47 kg/kg of fuel, 3.18 kg/kg of fuel]

1.17 The percentage composition by mass of a crude oil is given as follows: $C = 90\%$, $H_2 = 3.3\%$, $O_2 = 3\%$, $N_2 = 0.8\%$, $S = 0.9\%$, and the remaining is incombustible matter if 50% excess air is supplied, find the percentage of dry exhaust gases formed by volume.

[**Ans.** $CO_2 = 12.7\%$, $SO_2 = 0.05\%$, $O_2 = 7\%$, $N_2 = 80.25\%$]

1.18 The volumetric composition of the dry exhaust gases coming from a petrol engine is: $CO_2 = 12.4\%$, $CO = 3\%$, $N_2 = 84.6\%$. Assuming the fuel to be a pure hydrocarbon, find the ratio of carbon to hydrogen by mass in the fuel and the A:F ratio.

[**Ans.** C = 84.4%, H_2 = 15.6%, 14.1: 1]

ANSWERS TO MULTIPLE-CHOICE QUESTIONS

1. a
2. b
3. d
4. d
5. d
6. b
7. a
8. b
9. c
10. b
11. a
12. a
13. a
14. a
15. a
16. c
17. c
18. d
19. d
20. d
21. a
22. a
23. a
24. c
25. d
26. d
27. a
28. b
29. a
30. c

Chapter 2

Properties of Steam

2.1 ❐ PURE SUBSTANCE

A pure substance may be defined as a system that has the following characteristics:

1. **Homogeneous in composition:** It means that the composition of every part of the system is same throughout. By composition, we mean the relative proportions of the chemical elements into which the sample of the system can be analysed.
2. **Homogeneous in chemical aggregation:** It means that the chemical elements must be combined chemically in the same way in all parts of the system.
3. **Invariable in chemical aggregation:** It means that the state of chemical combination of the system does not change with time and position.

Examples of pure substance are atmospheric air, mixture of steam-water, combustion products of a fuel, etc.

2.2 ❐ CONSTANT PRESSURE FORMATION OF STEAM

Consider 1 kg of ice under atmospheric pressure and at a temperature of −10°C contained in a vessel. If we heat the ice gradually keeping the pressure constant, the following changes will take place, as shown in Fig. 2.1:

Figure 2.1 *Constant pressure temperature-heat added diagram for steam and water*

1. The temperature of ice will gradually increase till it just reaches the freezing temperature (represented by point '*b*'), which is 0°C at atmospheric pressure. This is shown by the straight line *ab*.
2. On addition of more heat at point *b*, the ice starts melting, and with no rise in temperature till the whole ice has melted, up to point *c*, the ice is converted into water. The heat added during the process *bc* is called the *latent heat of fusion of ice* or simply *latent heat of ice*.
3. Gradual addition of more heat beyond the point *c* leads to gradual increase in temperature till the water reaches the vapourisation or boiling point '*d*'. Under atmospheric pressure of 1.01325 bar, it is 100°C.
4. On further heating beyond point *d*, the water will gradually turn into steam, with no rise in temperature. This steam contains some water particles and is called the wet steam. With further addition of heat, all the water particles in the steam disappear and this steam is called dry saturated steam or simply dry steam. This corresponds to point '*e*'. The heat added during *de* is called the *latent heat of vapourisation of steam* or simply *latent heat of steam*.
5. If the dry saturated steam is further heated beyond point *e*, keeping the pressure constant, the process is called *superheating*. The steam so obtained is called the *superheated steam*. Let the steam be superheated to temperature t_s°C,

represented by point 'f'. Then $(t_s - t)$°C is called the *degree of superheat*. The heat added during process *ef* is called the *heat of superheat*.

2.3 ❒ PROPERTIES OF STEAM

1. **Enthalpy or total heat of water:** It is defined as the quantity of heat required to raise the temperature of 1 kg of water from 0°C to its boiling point or saturation temperature corresponding to the pressure applied. It is also called enthalpy of saturated water or liquid heat and is represented by h_f.

$$h_f = \text{specific heat of water, } c_{pw} \times \text{rise in temperature}$$
$$= 4.187 \times \Delta t, \text{ kJ/kg}$$

Note that enthalpy of saturated water can also be found from the steam table based on saturation temperature (Appendix A.1.1).

2. **Heat of steam or latent heat of vapourisation:** Latent heat of steam at a particular pressure may be defined as the quantity of heat required to convert 1 kg of water at its boiling point into dry saturated steam at the same pressure. It is denoted by h_{fg}.
3. **Dryness fraction or quality:** Dryness fraction of a liquid –vapour mixture is defined as the ratio of mass of dry steam actually present to the mass of the total mixture. It is denoted by x.

Let m_s = mass of dry steam (saturated vapour), kg

m_w = mass of water vapour in steam (saturated liquid), kg

x = dryness fraction of the sample

For dry steam, $m_w = 0$ and $x = 1$.

The *quality of steam* is the dryness fraction expressed as a percentage.

Wetness fraction = $1 - x$

4. **Specific volume of wet steam:** The specific volume of steam is the volume of 1 kg of dry steam.

Let V be the total volume of the liquid-vapour mixture, V_f be the volume of the saturated liquid and V_g be the volume of the saturated vapour.

Total volume of liquid and vapour can be expressed in terms of their specific volume as

$$V_f = m_w v_f$$
$$V_g = m_s v_g$$

where v_f and v_g are the specific volumes of saturated liquid and saturated vapour, respectively.

Specific volume of the mixture is then

$$= (1 - x)v_f + xv_g$$
$$= v_f + x(v_g - v_f)$$

Now $v_f << v_g$

5. **Enthalpy of wet steam:** It may be defined as the quantity of heat required to convert 1 kg of water at 0°C, at constant pressure, into wet steam. Similar to the analysis of specific volume, one can write

where h_f is the specific enthalpy of saturated water and h_{fg} is the change in specific enthalpy

during vapourisation. When steam is dry saturated, then $x = 1$

6. **Total enthalpy of superheated steam:**

   Let c_{ps} = specific heat of superheated steam

   t_s = temperature of formation of steam, °C

   t_{sup} = temperature of superheated steam, °C

   Then heat of superheat = $c_{ps}\,(t_{sup} - t_s)$

   Note that enthalpy of superheated steam can also be found from the steam table (Appendix A.1.3).

7. **Internal energy of steam:** The internal energy of steam may be defined as the actual heat energy stored in steam above the freezing point of water. It is the difference between the total heat of steam and the external work of evaporation. Similar to the analysis of specific volume, one can write for 1 kg of wet steam,

   where u_f is the specific internal energy of saturated water and u_{fg} is the change in specific internal energy during evaporation. From the known value of enthalpy, internal energy can be found as follows:

$$u_{wet} = h - pv_1$$

For superheated steam,

$$h_{\text{sup}} = h_g + c_{ps}\,(t_{\text{sup}} - t_s)$$
$$u_{sup} = h_{\text{sup}} - pv_{\text{sup}}$$

8. **Entropy of water:** Consider 1 kg of water to be heated from a temperature T_s to T at a constant pressure.

   Then, change of entropy =

   where c_{pw} = specific heat of water

   The entropy of water is always reckoned above the freezing point of water.

   Therefore, T_s = saturation temperature of formation of steam

   T = absolute temperature of freezing water = 273 K

   Entropy of saturated water,

9. **Entropy of wet steam:** Consider 1 kg of wet steam at an absolute pressure p bar.

   Let x = dryness fraction of steam

   T_s = temperature of formation of steam

   s_f = the specific enthalpy of saturated water

   s_{fg} = change in entropy during evaporation

   The entropy of wet steam can be calculated as

   When steam is dry saturated, then $x = 1$

10. **Entropy of superheated steam:** Consider 1 kg of dry

saturated steam at T_s temperature of formation, to be superheated to a temperature T_{sup}.

Let c_{ps} = specific heat of steam at constant pressure during superheating

Change of entropy per kg of steam during superheating

Total entropy of superheated steam

s_{sup} = Entropy of dry saturated steam + Change of entropy during superheating

2.4 ❐ STEAM TABLES

In actual practice, it is quite cumbersome to calculate the relation between various quantities such as pressure, temperature, volume, enthalpy, latent heat and entropy, etc., of steam. Such quantities have been determined experimentally and recorded in the form of tables known as *steam tables*. Steam table is a complete data book that

contains various properties, like volume, internal energy, enthalpy and entropy for 1 kg of water in different phases for a given pressure or temperature. Steam table is given in the Appendix A. The properties of steam are given in three different tables—saturated steam table based on temperature (Appendix A.1.1), saturated steam table based on pressure (Appendix A.1.2), and superheated steam table (Appendix A.1.3). When the temperature is selected as independent variable and the properties of steam are tabulated, the steam table is referred to as the saturated steam table based on temperature. When the pressure is chosen as independent variable, it is referred to saturated steam table based on pressure. For superheated steam,

there is a separate table for different amount of degree of superheat.

Values for unknown pressure or temperature can be calculated by linear interpolation between two consecutive values. Saturated steam table based on pressure for a particular pressure (= 1 bar) is shown in Table 2.1.

Table 2.1 *Saturated steam table based on pressure*

2.5 ❐ TEMPERATURE–ENTROPY DIAGRAM FOR WATER AND STEAM

It is the plot of the saturation temperature of water and steam corresponding to various absolute pressure vs entropies at those saturation temperatures, as shown in Fig. 2.2. The

curve on the left marked water line (saturated liquid line) shows the relationship between entropy and temperature before steam is formed. The dry steam line (saturated vapour line) shows the relationship between entropy and temperature when all the water has been converted into steam. The area in between two regions is called the *wet region*. The area to the right of the dry steam lines is called the region of superheat. The lines of constant dryness, constant volume, and constant total heat are also drawn on the diagram. The temperature at which the distinction between the entropy of water and that of dry steam disappears is called the *critical temperature* (374.15°C). The pressure corresponding to this

temperature is called the *critical pressure* (221.2 bar).

Figure 2.2 *Temperature−entropy diagram for water and steam*

2.6 ❐ ENTHALPY−ENTROPY OR MOLLIER DIAGRAM OF STEAM

In Mollier diagram (Fig. 2.3), the vertical ordinate represents the enthalpy and the horizontal ordinate represents the entropy. The regions above and below the saturation line represent the superheated and wet conditions of steam, respectively. The lines of constant dryness fraction are shown in the wet steam region, whereas the lines of constant temperature and pressure in the region of both wet and superheat are shown.

From the thermodynamic property

relationship, one can write,

$$Tds = dh - vdp$$

For constant pressure process, we have

$$Tds = dh$$

From Eq. (2.13), it is clear that the *slope of the constant pressure lines is equal to the corresponding saturation temperatures*. It may be noted that as the pressure increases, the saturation temperature also increases. This is the reason for which *constant pressure lines (isobar) are divergent* on the *h–s* diagram. The saturated liquid line and saturated vapour line meet at critical

point.

Figure 2.3 *Mollier diagram for steam*

2.7 ❐ VARIOUS PROCESSES FOR STEAM

2.7.1 Constant Volume Process

The constant volume heating process is shown on the p–v and T–s diagrams in Fig. 2.4. Consider 1 kg of wet steam at a certain pressure p_1, and dryness fraction x_1 is heated to state 2 in the wet region and then to state 3 in the superheated region.

Initial volume of steam $v_1 = v_{f1} + x_1(v_{g1} - v_{f1})$

Final volume of steam in wet region $v_2 = v_{f2} + x_2(v_{g2} - v_{f2})$

Since the volume remains constant,

$$v_{f1} + x_1(v_{g1} - v_{f1}) = v_{f2} + x_2(v_{g2} - v_{f2})$$

Figure 2.4 *Constant volume heating: (a) p−v diagram, (b) T−s diagram*

From steam table, internal energy of steam can be found as follows:

Initial internal energy of steam, $u_1 = u_{f1} + x_1 u_{fg1}$

Final internal energy of steam in the wet region, $u_2 = u_{f2} + x_2 u_{fg2}$

Since $dv = 0$, therefore $\delta W = 0$

For constant volume heating process, $v_2 = v_3$. After knowing the value of v_3, the condition of steam can be found from

superheated steam table. If pressure of state 3 is given ($=p_3$), then from known values of p_3 and v_3, temperature of state 3 can be obtained. Similarly, if temperature at state 3 ($=T_3$) is given, pressure can be found from known values of T_3 and v_3.

2.7.2 Constant Pressure Process

The constant pressure heating process is shown in Fig. 2.5. In this case, $p_1 = p_2 = p$ (say).

For wet steam, during heating process 1–2,

Work done per kg of steam, $w = pdv = p(v_2 - v_1)$

where $v_1 = v_f + x_1(v_g - v_f)$ and $v_2 = v_f +$

$x_2(v_g - v_f)$

The specific enthalpy of steam, $h_1 = h_f + x_1 h_{fg}$ and $h_2 = h_f + x_2 h_{fg}$

From first law of thermodynamics, heat transfer per kg of steam during constant pressure heating

$$q = h_2 - h_1$$
$$q = (h_f + x_2 h_{fg}) - (h_f + x_1 h_{fg})$$

For superheated steam, specific volume and specific enthalpy of steam can be found from the superheated steam table. Then, during heating process 2-3,

Work done per kg of steam, $w = pdv = p(v_3 - v_2) = p(v_{sup} - v_2)$

and heat transfer per kg of steam

Figure 2.5 *Constant pressure expansion*

2.7.3 Isothermal Process

The isothermal heating process is shown in Fig. 2.6. The isothermal process for wet steam will be same as the constant pressure process. However, in the superheated region, the steam will behave as a gas and shall follow Boyle's law, pv = const. Therefore, the process will be hyperbolic.

In the wet region,

$$q = h_2 - h_1 \text{ and } w = p_1 (v_2 - v_1)$$

In the superheated region,

$q = h_3 - h_2$ and $w = p_1 (v_3 - v_2)$

Figure 2.6 *Isothermal process*

2.7.4 Hyperbolic Process

During a hyperbolic process, pv = const. If the superheated vapour behaves as an ideal gas, then the process during which pv = const. may be regarded isothermal, as shown in Fig. 2.7.

For 1 kg of steam in the wet region,

Initial volume of steam $v_1 = v_{f1} + x_1(v_{g1} - v_{f1})$

Final volume of steam $v_2 = v_{f2} + x_2(v_{g2} - v_{f2})$

As pv = const., therefore

or

Figure 2.7 *Hyperbolic expansion*

Now, $h_1 = h_{f1} + x_1 h_{fg1}$ and $h_2 = h_{f2} + x_2 h_{fg2}$ for wet steam

and $h_{s2} = h_{g2} + c_p\,(T_{sup} - T_2)$ for superheated steam

Change is internal energy,

where v_1 and v_2 are the specific volumes of steam before and after heating.

Total heat supplied, $q = w + (u_2 - u_1)$

2.7.5 Reversible Adiabatic or Isentropic Process

The isentropic process of expansion

(1-2) is shown in Fig. 2.8 on *T*–*s* diagram. The initial condition of steam may be wet, dry, or superheated. For initially wet steam as shown in Fig. 2.8(a), we have

$$s_1 = s_{f1} + x_1 s_{fg1}$$
$$s_2 = s_{f2} + x_2 s_{fg2}$$

Since $s_1 = s_2$, therefore

$$s_{f1} + x_1 s_{fg1} = s_{f2} + x_2 s_{fg2}$$

For initially dry steam (Fig. 2.8(b)), $s_1 = s_{g1}$

Figure 2.8 *Isentropic expansion: (a) Expanded steam wet, (b) Expanded steam dry saturated, (c) Expanded steam superheated*

$$s_2 = s_{f2} + x_2 s_{fg2}$$

Then, $s_{g1} = s_{f2} + x_2 s_{fg2}$

For initially superheated steam at temperature T_{sup}, s_1 can be calculated from superheated steam table. After expansion, state 2 may be superheated (Fig. 2.8(c)), dry or wet. If the state after expansion is superheated, then the condition of steam can be calculated from the relationship $s_1 = s_2$. Note that if pressure is specified, then temperature can be calculated or otherwise if temperature is specified, then pressure can be calculated.

Now $q = w + (u_2 - u_1)$

where $u_1 = u_{f1} + x_1 u_{fg1}$ for wet steam

$= u_{g1}$ for dry steam

$= u_{sup}$ for superheated steam

Similarly, u_2 can be found depending on the condition of steam after expansion.

But $q = 0$

$$w = u_1 - u_2$$

2.7.6 Polytropic Process

Let the steam expand from condition 1 to 2 according to the law pv^n = constant. Then for wet steam,

Initial volume of steam, $v_1 = v_{f1} + x_1(v_{g1} - v_{f1})$

Final volume of steam $v_2 = v_{f2} + x_2(v_{g2} - v_{f2})$

or

or

Also $u_1 = u_{f1} + x_1 u_{fg1}$

$$u_2 = u_{f2} + x_2 u_{fg2}$$

$n = 1.13$ for wet steam and 1.3 for superheated steam

For superheated steam on heating from wet steam, v_2 can be calculated from superheated steam table for the known pressure and temperature conditions. Similarly, the condition of steam can also be found after knowing the value of v_2 from Eq. (2.21).

2.7.7 Throttling Process

The throttling process occurs when steam is expanded through a small aperture as in the case of throat of a nozzle. During this process, no work is done and the enthalpy remains constant. The throttling process is shown on T–s and h–s diagrams in Fig. 2.9.

$$h_1 = h_{f1} + x_1 h_{fg1}$$
$$h_2 = h_{f2} + x_2 h_{fg2} \text{ for wet steam after expansion}$$
$$= h_{f2} + h_{fg2} + c_{ps} (T_{\text{sup}} - T_s), \text{ for superheated steam after expansion}$$

Now $h_1 = h_2$

For wet steam after expansion

For superheated steam after expansion

Figure 2.9 *Throttling process: (a) T−s diagram, (b) h−s diagram*

2.8 ❐ DETERMINATION OF DRYNESS FRACTION OF

The different methods commonly used for the measurement of dryness fraction of steam are as follows:

1. Barrel Calorimeter
2. Separating Calorimeter
3. Throttling Calorimeter
4. Combined Separating and Throttling Calorimeter.

The working principle of all the calorimeters is to bring the state of the substance from the two-phase region to the single phase region (either compressed liquid or superheated vapour region). This is because in the two-phase region, pressure and temperature are not independent variable, one is independent and the other is dependent. On the other hand, both in compressed liquid and superheated vapour region, both

pressure and temperature are independent variables.

2.8.1 Barrel Calorimeter

The barrel calorimeter, shown in Fig. 2.10, consists of a copper barrel placed on wooden block and covered by a wooden cover. The barrel contains a known quantity of cold water and is surrounded by an outer vessel with air space in between, which acts as an insulator. The temperature is measured by a thermometer that passes through one of the holes in the wooden cover. The whole assembly is placed on the platform of a weighing bridge. The steam from the main steam pipe enters through the sampling tube via the control valve and flows into the cold

water through fine exit holes of the ring provided at the end of the pipe. The condensation of steam takes place as it comes in contact with cold water, and as a result, the temperature of water rises. The quantity of steam condensed can be known from the difference of readings of the weighing bridge before and after the condensation of steam. The pressure of entering steam is indicated by the pressure gauge.

Figure 2.10 *Barrel calorimeter*

Let m_s = mass of steam condensed

x = dryness fraction of steam

h_{fg} = latent heat of steam

t_s = temperature of formation of steam

(corresponding to pressure indicated by the pressure gauge)

m_w = mass of cold water in the calorimeter in the beginning

m_c = water equivalent of calorimeter

t_1 = initial temperature of cold water in the calorimeter

t_2 = final temperature of the mixture after steam condensation

Heat lost by steam = $m_s[x \times h_{fg} + c_{ps}(t_s - t_2)]$

Heat gained by cold water and calorimeter = $(m_w + m_c) \times c_{pw} \times (t_2 - t_1)$

Heat lost by steam = Heat gained by cold water and calorimeter

This method gives only approximate results. The value of x calculated is always lower than the actual value as heat losses due to convection and radiation are not accounted for.

2.8.2 Separating Calorimeter

This calorimeter, shown in Fig. 2.11, consists of two concentric chambers that communicate with each other through an opening at the top. The steam to be tested flows from the main steam pipe through a sampling tube to the calorimeter pipe through the valve which must be fully open when testing. The metal basket has a large number of

perforations through which the steam discharges. The water particles due to their heavier momentum get separated from the steam and get collected in the inner chamber. The quantity of water collected is indicated by the level in the gauge glass and the pointer that moves on a scale. The dry steam in the inner chamber moves up and then down again through the annular space between the two chambers. The gauge fitted has two scales: the inner one indicates the pressure, whereas the outer one indicates the rate of discharge of dry saturated steam during a predefined time interval. Then the steam then flows through the calibrated orifice to the bucket calorimeter which is placed on the platform balance by which the most

of steam discharged can be further confirmed.

Figure 2.11 *Separating calorimeter*

Let m_s = mass of dry steam condensed

m_w = mass of suspended moisture collected

It is not possible to remove all the water particles from steam, and the dryness fraction calculated by this method is always greater than the actual. However, it is a quick method and is used for the measurement of dryness fraction for very wet steam.

2.8.3 Throttling Calorimeter

The throttling calorimeter, shown in Fig.

2.12, consists of the sampling tube through which the steam from the main steam pipe flows through the throttling valve and becomes superheated. The steam then flows into the inner chamber, flows down, and rises up again to enter the annular space. The loss of heat by radiation from the inner chamber is minimised by the hot steam around the outside of the inner chamber. The temperature of throttled steam is measured by the thermometer placed in the pocket filled with cylinder oil. To obtain good results, the steam issuing from the throttle valve must be superheated. To ensure this, a manometer is attached to measure the pressure of steam. Corresponding to this pressure, the saturation temperature of

steam must be lower than the temperature indicated by the thermometer. The steam finally escapes through the exhaust. This calorimeter is suitable for steam having high dryness fraction.

Let p_1 = initial pressure of wet steam before expansion

x_1 = dryness fraction of wet steam

h_{fg1} = latent heat of vapourization of wet steam

t_{sup} = temperature of superheated steam after expansion

h_{fg2} = latent heat of vapourization of steam after expansion

t_{s2} = saturation temperature of steam after expansion

c_{ps} = specific heat of superheated steam at constant pressure

Enthalpy of steam before throttling = $h_{f1} + x_1 h_{fg1}$

Figure 2.12 *Throttling calorimeter*

Enthalpy of steam after throttling = $h_{f2} + h_{fg2} + c_{ps} (t_{\text{sup}} - t_{s2})$

Enthalpy before throttling = Enthalpy after throttling

$$h_{f1} + x_1 h_{fg1} = h_{f2} + h_{fg2} + c_{ps} (t_{\text{sup}} - t_{s2})$$

2.8.4 Combined Separating and Throttling Calorimeter

The combined separating and throttling calorimeter is shown in Fig. 2.13. The steam is first passed through the separating calorimeter where it losses most of its moisture and becomes comparatively drier. It is then passed through the throttling calorimeter where superheating takes place without change of enthalpy. The temperature and pressure of steam after throttling are measured by using a thermometer and pressure gauge, respectively.

Let p_1 = pressure of wet steam entering the throttling calorimeter

x_1 = dryness fraction of steam

h_{fg1} = latent heat of entering steam

m_w = mass of suspended moisture collected

m_s = mass of steam leaving the separating calorimeter and entering the throttling calorimeter

p_2 = pressure of steam after throttling

h_{fg2} = latent heat of steam at pressure p_2

Enthalpy of steam entering throttling calorimeter = Enthalpy of steam leaving throttling calorimeter

$$h_{f1} + x_1 h_{fg1} = (h_{f2} + h_{fg2}) + c_{ps}\,(t_{\text{sup}} - t_{s2})$$

Figure 2.13 *Combined separating and throttling calorimeter*

and x = dryness fraction of steam entering the separating calorimeter

Mass of dry steam entering the whole apparatus = $m_s\, x_1$

This calorimeter gives quite satisfactory results when the steam is considerably wet.

Example 2.1

Determine the condition of steam in the following cases:

1. At a pressure of 10 bar and temperature 200°C.
2. At a pressure of 10 bar and specific volume 0.175 m^3/kg.

Solution

Given p = 10 bar; t = 200°C; v =

$0.175\ m^3/kg$

1. ***Condition of steam at temperature of 200°C:***

   From steam table (A.1.2 in the Appendix), corresponding to a pressure of 10 bar, we find the saturation temperature $t_s = 179.91°C$

   Since the saturation temperature at 10 bar is (179.91°C) lower than the given temperature of the steam (200°C), therefore the given steam is superheated.

   The degree of superheat = 200 − 179.91= 20.09°C

2. **Condition of steam at a volume of 0.175 m^3/kg:**

   From steam table (A.1.2 in the Appendix), $v_f = 0.001127\ m^3/kg$, $v_g = 0.19444\ m^3/kg$.

   Since the volume of given steam ($0.175\ m^3/kg$) is less than the specific volume of the dry saturated steam ($0.19444\ m^3/kg$), therefore the given steam is wet. Therefore, the dryness fraction can be found to be

$$0.175 = 0.001127 + x\,(0.19444 - 0.001127)$$

or $x = 0.8994$

Example 2.2

Calculate the quantity of heat required to generate 1 kg of steam at a pressure of 8 bar from water at 30°C (a) when dryness fraction is 0.9, (b) when steam is just dry, and (c) when the temperature of steam is 300°C.

Solution

From steam tables (A.1.2 in the appendix) at 8 bar: $h_f = 721.1$ kJ/kg, $h_{fg} = 2048.0$ kJ/kg

And at 30°C (A.1.1 in the appendix), $h_f = 125.79$ kJ/kg

1. $h = h_f + x\, h_{fg} = 721.1 + 0.9 \times 2048.0 = 2564.3$ kJ/kg

Net heat required = $h - h_f$ = 2564.3 − 125.79 = 2438.51 kJ/kg

2. $h = h_f + h_{fg} = h_g$ = 2769.1 kJ/kg

Net heat required = $h - h_f$ = 2769.1 − 125.79 = 2643.31 kJ/kg

3. t_s = 170.4°C, t_{sup} = 300°C

Therefore, the steam is in superheated condition. From superheated steam table (A.1.3 in the appendix), we have h = 3056.4 kJ/kg

Net heat required = $h - h_f$ = 3056.4 − 125.77 = 2930.63 kJ/kg

Example 2.3

A pressure cooker contains 2 kg of steam at 5 bar and 0.9 dry. Find the

quantity of heat that must be rejected so that the quality of steam becomes 0.5 dry.

Solution

The pressure cooker is a constant volume vessel. Therefore, $W = 0$

$$\therefore Q = \Delta U = U_2 - U_1$$

From steam tables (A.1.2 in the appendix) at 5 bar: $v_f = 0.001093$ m^3/kg, $v_g = 0.3749$ m^3/kg, $u_f =$ 639.66 kJ/kg, u_{fg} = 1921.6 kJ/kg,

$$u_1 = u_f + x_1 u_{fg} = 639.66 + 0.9 \times 1921.6 = 2369.1 \text{ kJ/kg}$$
$$U_1 = 2 \times 2369.1 = 4738.2 \text{ kJ}$$

Specific volume of the steam at 5 bar

$$v_1 = v_f + x_1 (v_g - v_f) = 0.001093 + 0.9 \times (0.3749 - 0.001093) = 0.3375 \text{ m}^3/\text{kg}$$

From steam tables (A.1.2 in the appendix), the properties of steam at 2.5 and 2.75 bar are given below:

For constant volume of steam, we have

$$v_1 = v_2 = 0.3375 \text{ m}^3/\text{kg}$$

Specific volume of the steam at 2.5 bar

$v = v_f + x\,(v_g - v_f) = 0.001067 + 0.9 \times (0.7187 - 0.001067) = 0.3599\ m^3/kg$

Specific volume of the steam at 2.75 bar

$v = v_f + x(v_g - v_f) = 0.001070 + 0.9 \times (0.6573 - 0.001070) = 0.3292\ m^3/kg$

Using linear interpolation, the pressure corresponding to $v_2 = 0.3375\ m^3/kg$ is obtained as

or $p_2 = 2.68$ bar

Using linear interpolation, we get

or $u_{fg2} = 1994.83$ kJ/kg

$\therefore u_2 = u_{f2} + x_2\, u_{fg2} = 544.79 + 0.5 \times 1994.83 = 1542.205$ kJ/kg

$$U_2 = m\, u_2 = 2 \times 1542.205 = 3084.41 \text{ kJ}$$
$$Q = U_2 - U_1 = 3084.41 - 4738.2 = -1653.79 \text{ kJ}$$

Example 2.4

About 5400 m^3/h of wet steam with dryness fraction 0.92 and at 10 bar is to be supplied by a boiler for a processing plant. Calculate (a) the mass of steam supplied per hour and (b) the quantity of coal of calorific value 15 MJ/kg to be burnt in the boiler if overall efficiency of the boiler is 30%.

Solution

1. From steam tables (A.1.2 in the appendix) at 10 bar: v_f = 0.001127 m³/kg, v_g= 0.19444 m³/kg

   Specific volume of the steam $v = v_f + x\,(v_g - v_f) = 0.001127 + 0.92 \times (0.19444 - 0.001127) = 0.179$ m³/kg

2. At p = 10 bar, h_f = 762.79 kJ/kg, h_{fg} = 2015.3 kJ/kg

   $h = h_f + x\,h_{fg} = 762.79 + 0.92 \times 2015.3 = 2616.866$ kJ/kg

   $H = \dot{m}h = 30167.6 \times 2616.866 = 78944566$ kJ/h

   Heat supplied to boiler, H_b = = 263148553 kJ/h

   Quantity of coal burnt = = 17.54 tonnes/h

Example 2.5

Find the internal energy of 1 kg of superheated steam at a pressure of 10 bar and 300°C. If this steam is expanded to 1.5 bar and 0.9 dryness,

then find the change in internal energy.

Solution

From steam table (Appendix A.1.3), at p = 10 bar and t_{sup} = 300°C, for superheated steam,

$$u_1 = 2793.2 \text{ kJ/kg}$$

For dry saturated steam, at p = 1.5 bar, we have from table A.1.2 in the Appendix

$$u_f = 466.92 \text{ kJ/kg}, \; u_{fg} = 2052.7 \text{ kJ/kg},$$
$$u_2 = u_{f2} + x_2 u_{fg2} = 466.92 + 0.9 \times 2052.7 = 2314.35 \text{ kJ/kg}$$
$$\Delta u = u_2 - u_1 = 2314.35 - 2793.2 = -478.85 \text{ kJ/kg}$$

Example 2.6

A rigid vessel is initially divided into two parts *A* and *B* by a thin partition. Part *A* contains 1 kg of steam at 4 bar, dry and saturated, and part *B* contains 2 kg of steam at 8 bar with dryness fraction of 0.90. The partition is removed and the pressure in the vessel after sometimes is found to be 6 bar. Find (a) the volume of the vessel and the state of steam and (b) the amount of heat transfer from steam to the vessel and the surroundings.

Solution

1. At $p_A = 4$ bar, from steam table A.1.2 in the Appendix, $v_{gA} = 0.4625$ m^3/kg, $u_A = 2553.6$ kJ/kg

$$V_A = m_A v_{gA} = 1 \times 0.4625 = 0.4625 \text{ m}^3$$

At $p_B = 8$ bar, from steam table A.1.2 in the Appendix, $v_{fB} = 0.001115$ m^3/kg, $v_{gB} = 0.2404$ m^3/kg

$$V_B = m_B (v_{fB} + x_B v_{gB}) = 2 \times [0.001115 + 0.9 \times (0.2404 - 0.001115)] = 0.433 \text{ m}^3$$

Volume of vessel, $V = V_A + V_B = 0.4625 + 0.433 = 0.8955$ m^3

At $p = 6$ bar, from steam table A.1.2 in the Appendix, $v_f = 0.001101$ m^3/kg, $v_g = 0.3157$ m^3/kg,

$$m = 1 + 2 = 3 \text{ kg}$$

Dryness fraction at 6 bar, x can be found to be

or $x = 0.945$

2. Now $U_A = m_A u_A = 1 \times 2553.6 = 2553.6$ kJ

$$U_B = m_B (h_{fB} + x_B h_{fgB} - p_B x_B v_{gB} \times 10^2)$$
$$= 2 (721.1 + 0.9 \times 2048.0 - 8 \times 0.9 \times 0.240 \times 10^2)$$
$$= 4783.0 \text{ kJ}$$

At 6 bar, $U = m[h_f + x\, h_{fg} - p x v_g \times 10^2]$

$$= 3[670.6 + 0.944 \times 2086.3 - 6 \times 0.944 \times 0.316 \times 10^2)$$
$$= 7383.25 \text{ kJ}$$
$$Q = U - (U_A + U_B)$$
$$= 7383.25 - (2553.3 + 4783.0) = 46.95 \text{ kJ}$$

Example 2.7

Calculate enthalpy and entropy for 5 kg of steam at 8 bar under the following conditions:

(a) dry and saturated, (b) wet steam having wetness of 40%, and (c) superheated steam at 250°C.

Solution

From steam table (Appendix A.1.2), at p = 8 bar, we have

h_f = 721.1 kJ/kg, h_{fg} = 2048.0 kJ/kg, h_g = 2769.1 kJ/kg, s_f = 2.0461 kJ/kgK, s_{fg} = 4.6166 kJ/kgK, s_g = 6.6627 kJ/kgK

At t_{sup} = 250°C and p = 8 bar, we have from table A.1.3

$h = 2950.0$ kJ/kg, $s = 7.0384$ kJ/kgK

1. $H = m\,h_g = 5 \times 2769.1 = 13845.5$ kJ

   $S = m\,s_g = 5 \times 6.6627 = 33.3135$ kJ/K

2. $x = 1 - 0.4 = 0.6$

   $H = m\,(h_f + x\,h_{fg}) = 5(721.1 + 0.6 \times 2048.0) =$ 9749.5 kJ

   $S = m\,(s_f + xs_{fg}) = 5(2.0461 + 0.6 \times 4.6166) =$ 24.08 kJ/K

3. $H = mh = 5 \times 2950.0 = 14750.0$ kJ

   $S = ms = 5 \times 7.0384 = 35.192$ kJ/K

Example 2.8

About 1 kg of steam 0.85 dry at a pressure of 1 bar and dryness fraction 0.85 is compressed in a cylinder to a pressure of 2 bar according to $pv^{1.25}$ = const. Determine the final condition of steam and the heat transfer through

the cylinder walls.

Solution

From steam table (Appendix A.1.2), at $p = 1$ bar, $v_{f1} = 0.001043$ m^3/kg, $v_{g1} = 1.694$ m^3/kg

$$u_{f1} = 417.33 \text{ kJ/kg}, u_{fg1} = 2088.7 \text{ kJ/kg}$$

Specific volume of steam

$$v = v_f + x (v_g - v_f) = 0.001043 + 0.85 \times (1.694 - 0.001043) = 1.44 \text{ m}^3/\text{kg}$$

Now $p_1 v_1 n = p_2 v_2 n$

At $p_2 = 2$ bar, $v_{f2} = 0.001061$ m^3/kg, $v_{g2} = 0.8857$ m^3/kg, $u_{f2} = 504.47$ kJ/kg, $u_{fg2} = 2025$ kJ/kg

Now $v_2 = 0.8271 = v_{f2} + x_2 (v_{g2} - v_f$

$_2) = 0.001061 + x_2\ (0.8857 - 0.001061)$

or $x_2 = 0.934$

Now $u_1 = u_{f1} + x_1 u_{fg1} = 417.33 + 0.85 \times 2088.7 = 2192.72$ kJ/kg

$u_2 = u_{f2} + x_2 u_{fg2} = 504.47 + 0.934 \times 2025 = 2395.82$ kJ/kg

$\Delta u = u_2 - u_1 = 2395.82 - 2192.72 = 203.1$ kJ/kg

Work done,

Heat transfer through the cylinder walls,

$$q = w + (u_2 - u_1) = -85.68 + 203.1 = 117.42 \text{ kJ/kg}$$

Example 2.9

Steam at a pressure of 10 bar and 0.9 dryness fraction expands to 1 bar hyperbolically. Find (a) work done and heat absorbed and (b) internal energy and change in enthalpy.

Take specific heat of steam at constant pressure = 2 kJ/kg.K.

Solution

From steam table (Appendix A.1.2), we have

$$v_1 = v_{f1} + x_1 (v_{g1} - v_{f1}) = 0.001127 + 0.9 \times (0.19444 - 0.001127) = 0.1751 \text{ m}^3/\text{kg}$$

Now, $p_1 v_1 = p_2 v_2$

Expansion ratio,

Work done, $w = p_1 v_1 \ln r = 10 \times 10^2 \times 0.1751 \ln 10 = 403.18$ kJ/kg

$$v_2 = 10 v_1 = 10 \times 0.1751 = 1.751 \text{ m}^3/\text{kg}$$

Now, $v_2 > v_{g2}$. Therefore, the steam is superheated at $p_2 = 1$ bar.

Using linear interpolation, we have

$$T_{\text{sup } 2} = 111.47°\text{C}$$

$h_1 = h_{f1} + x_1 h_{fg1} = 762.79 + 0.9 \times 2015.3 = 2576.56$ kJ/kg

Using linear interpolation, we get

$h_2 = 2699.19$ kJ/kg

$\Delta h = h_2 - h_1 = 2699.19 - 2576.56 =$

122.63 kJ/kg

$\Delta u = \Delta h$ for hyperbolic process

$q = w + \Delta u = 403.18 + 122.63 =$ 525.81 kJ/kg

Example 2.10

Steam, which is initially dry saturated, expands isentropically from pressure of 15 bar to 0.15 bar. Find the index of isentropic expansion.

Solution

From steam table (Appendix A.1.2), we have

For isentropic expansion, $s_1 = s_2$

$$s_1 = s_{f1} + s_{fg1} = 2.315 + 4.130 = 6.445 \text{ kJ/kg.K}$$
$$s_2 = s_{f2} + x_2 s_{fg2} = 0.7548 + x_2 \times 7.2536 = 6.445$$

or $x_2 = 0.7845$

$$v_1 = v_{g1} = 0.13177 \text{ m}^3\text{/kg}$$
$$v_2 = v_{f2} + x_2 (v_{g2} - v_{f2}) = 0.001014 + 0.7845 \times (10.022 - 0.001014) = 7.8625 \text{ m}^3\text{/kg}$$

Now, $p_1 v_1^n = p_2 v_2^n$

or $n = 1.126$

Example 2.11

Steam at a pressure of 8 bar and 0.95 dry is expanded to a pressure of 1.8 bar. Find the final condition of steam and the heat drop for each of the following methods of

expansion:

1. Adiabatic expansion
2. Throttling expansion

Solution

Given that $p_1 = 8$ bar, $x_1 = 0.95$, $p_2 = 1.75$ bar

From steam tables (Appendix A.1.2), at 8 bar, $s_{f1} = 2.0461$ kJ/kg.K, $s_{fg1} = 4.6166$ kJ/kg.K, $h_{f1} = 721.1$ kJ/kg, $h_{fg1} = 2048$ kJ/kg, and at 1.75 bar, $s_{f2} = 1.4848$ kJ/kg.K, $s_{fg2} = 5.6868$ kJ/kg.K, $h_{f2} = 486.97$ kJ/kg, $h_{fg2} = 2213.6$ kJ/kg

1. During an adiabatic expansion, $s_1 = s_2$

$$s_{f1} + x_1 s_{fg1} = s_{f2} + x_2 s_{fg2}$$

$$2.0461 + 0.95 \times 4.6166 = 1.4848 + x_2 \times 5.6868$$

$$x_2 = 0.8699$$

2. During throttling process, $h_1 = h_2$

$$h_{f1} + x_1 h_{fg1} = h_{f2} + x_2 h_{fg2}$$

$$721.1 + 0.95 \times 2048.0 = 486.97 + x_2 \times 2213.6$$

$$x_2 = 0.9847$$

Example 2.12

Steam at 10 bar flows into a barrel containing 150 kg of water at 10°C. The final temperature of water is 25°C and the increase in mass is 4 kg. Calculate the dryness fraction of steam. Take c_{pw} = 4.187 kJ/kg.K.

Solution

Given that m_s = 4 kg, p = 10 bar, m_w = 150 kg, t_{w1} = 10°C, t_{w2} = 25°C, and c_{ps} = 4.187 kJ/kg.K

From steam table (Appendix A.1.2), at 10 bar, h_{fg} = 2015.3 kJ/kg and t_s = 179.91°C

Heat lost by steam = Heat gained by water

$$m_s[x\,h_{fg} + c_{pw}(t_s - t_{w2})] = m_w\,(t_{w2} - t_{w1})c_{pw}$$
$$4[x \times 2015.3 + 4.187(179.91 - 25)] = 150\,(25 - 10) \times 4.187$$

Dryness fraction of steam, $x = 0.847$

Example 2.13

In a separating and throttling calorimeter, the steam main pressure is 8 bar absolute and the temperature after throttling is 130°C. The pressure in the throttling calorimeter is 0.001 bar gauge, and the barometer reading is 75 cm. If 0.15 kg of water is trapped in the separator and 1.8 kg of steam is passed through the throttling

calorimeter, then determine the dryness fraction of steam in the steam main.

Solution

After throttling, $p_2 = 0.001 + 0.75 \times 9.81 \times 13600 \times 10^{-5} = 1.01$ bar

The saturation temperature for $p_2 = 1.01$ bar, $t_{s2} = 99.62°C$

Degree of superheat $= t_{sup} - t_{s2} = 130 - 99.62 = 30.38°C$

For superheated steam, from steam tables (Appendix A.1.3), for $t_{sup} = 130°C$

$p = 1.01$ bar, we have by linear interpolation

$$h_{sup2} = 2736.32 \text{ kJ/kg}$$

From steam tables (Appendix A.1.2),

At 8 bar h_{f1} = 721.1 kJ/kg, h_{fg1} = 2048 kJ/kg

$$h_1 = h_{f\,1} + x_1 h_{fg1}$$

Now $h_1 = h_{\text{sup}2}$

$$721.1 + x_1 + 2048.0 = 2736.32$$
$$x_1 = 0.984$$

Dryness fraction, $x = x_1 x_2$

$$= 0.984 \times 0.923 = 0.908$$

Example 2.14

Calculate the enthalpy of 1 kg of steam at a pressure of 8 bar and dryness fraction of 0.8. How much heat would be required to raise 2 kg

of this steam from water at 20°C?

Solution

Given that $m_s = 1$ kg, $p = 8$ bar, $x = 0.8$, and $t_w = 20$°C

Enthalpy of 1 kg of steam:

From steam table (Appendix A.1.2), corresponding to a pressure of 8 bar, we find that,

$$h_f = 721.1 \text{ kJ/kg and } h_{fg} = 2048 \text{ kJ/kg}$$

Enthalpy of 1 kg of wet steam,

$$h = h_f + x\, h_{fg} = 721.1 + 0.8 \times 2048 = 2359.5 \text{ kJ}$$

Heat required to raise 2 kg of this steam from water at 20°C:

We have calculated above the enthalpy or total heat required to raise 1 kg of steam from water at 0°C. Since the water, in this case, is already at 20°C, therefore,

Heat already in water = $4.2 \times 20 = 84$ kJ

Heat required per kg of steam = $2359.5 - 84 = 2275.5$ kJ

and heat required for 2 kg of steam $= 2 \times 2275.5 = 4551$ kJ

Example 2.15

Determine the quantity of heat required to produce 1 kg of steam at

a pressure of 6 bar at a temperature of 25°C, under the following conditions:

1. When the steam is wet having a dryness fraction 0.9
2. When the steam is dry saturated
3. When it is superheated at a constant pressure at 250°C, assuming the mean specific heat of superheated steam to be 2.3 kJ/kg.K.

Solution

Given that p = 6 bar, t_w = 25°C, x = 0.9, t_{sup} = 250°C, and c_{ps} = 2.3 kJ/kg.K

From steam table (Appendix A.1.2), corresponding to a pressure of 6 bar, we find that

h_f = 670.4 kJ/kg, h_{fg} = 2086.3 kJ/kg, and t_s = 158.85°C

1. When the steam is wet:

   Enthalpy or total heat of 1 kg of wet steam,

   $h = h_f + xh_{fg}$ = 670.54 + 0.9 × 2086.3 = 2548.21

kJ

Since the water is at a temperature of 25°C, therefore from steam tables (Appendix A.1.1)

Enthalpy of water 104.87 kJ

∴ Heat actually required = 2548.21 – 104.87 = 2443.34 kJ

2. When the steam is dry saturated:

From steam tables (Appendix A.1.2), enthalpy or total heat of 1 kg of dry saturated steam,

$$h_g = 2756.8 \text{ kJ}$$

∴ Heat actually required = 2756.8 – 104.87 = 2651.93 kJ

3. When the steam is superheated:

From steam tables (Appendix A.1.3), enthalpy or total heat of 1 kg of superheated steam,

$$h_{sup} = 2957.2 \text{ kJ}$$

∴ Heat actually required = 2957.2 – 104.87 = 2852.33 kJ

Example 2.16

Steam enters an engine at a pressure of 12 bar with a 67°C of superheat. It is exhausted at a pressure of 0.15 bar and 0.95 dry. Find the drop in enthalpy of the steam. Assume specific heat of superheated steam c_{ps} = 2.1 kJ/kgK.

Solution

Given that p_1 = 12 bar, $t_{\text{sup}} - t_s$ = 67°C, p_2 = 0.15 bar, and x = 0.95

From steam table (Appendix A.1.2), corresponding to a pressure of 12 bar, we find that

$$h_f = 798.64 \text{ kJ/kg}; \; h_{fg} = 1986.2 \text{ kJ/kg}; \; h_g = 2784.8 \text{ kJ/kg}$$

Enthalpy or total heat of 1 kg of superheated steam,

$$h_{sup} = h_g + c_{ps}\,(t_{sup} - t_s)$$
$$= 2784.8 + 2 \times 67 = 2925.5 \text{ kJ/kg}$$

Similarly, from steam tables (Appendix A.1.2), corresponding to a pressure of 0.15 bar

$$h_f = 225.91 \text{ kJ/kg};\ h_{fg} = 2373.1 \text{ kJ/kg}$$

Enthalpy or total heat of 1 kg of wet steam,

$$h = h_f + xh_{fg} = 225.91 + 0.95 \times 2373.1 = 2480.36 \text{ kJ/kg}$$

∴ Drop in enthalpy of the steam = $h_{sup} - h = 2925.5 - 2480.36 = 445.14$ kJ/kg

Example 2.17

A steam engine obtains steam from a boiler at a pressure of 15 bar and

0.98 dry. It was observed that the steam loses 21 kJ of heat per kg as it flows through the pipe line and pressure remaining constant. Calculate the dryness fraction of the steam at the engine end of the pipeline.

Solution

Given that p = 15 bar, x = 0.98, and heat loss = 21 kJ/kg

From steam table A.1.2 in the Appendix, corresponding to a pressure of 15 bar we find that

h_f = 844.87 kJ/kg; h_{fg} = 1947.3 kJ/kg

Enthalpy of wet steam at the boiler end.

$$h = h_f + xh_{fg} = 844.87 + 0.98 \times 1947.3 = 2753.22 \text{ kJ/kg}$$

Since the steam loses 21 kJ/kg of steam, therefore enthalpy of wet steam at the engine end,

$$h_2 = 2753.22 - 21 = 27332.22 \text{ kJ}$$

Let x_2 = Dryness fraction of steam at the engine end.

Since the pressure remains constant, therefore h_f and h_{fg} is same. We know that

$$h_2 = h_f + x_2 h_{fg}$$
$$2732.22 = 844.87 + x_2 \times 1947.3$$

or $x_2 = 0.9692$

Example 2.18

A coal fired boiler plant consumes 400 kg of coal per hour. The boiler evaporates 3200 kg of water at 45°C into superheated steam at a pressure of 12 bar and 274.5°C. If the calorific value of the fuel is 32600 kJ/kg of coal, then determine the following:

1. Equivalent evaporation
2. Efficiency of boiler

Assume specific heat of superheated steam as 2.1 kJ/kg.K.

Solution

Given that $m_f = 400$ kg/h, $m_s = 3200$ kg/h, feed water temperature, $t_1 =$ 45°C, $p = 12$ bar,

$T_{\text{sup}} = 274.5°\text{C}$, C.V. = 32600 kJ/kg, $c_{ps} = 2.1$ kJ/kg.K

From steam table A.1.2 in the Appendix, at 12 bar, we have

$T_s = 187.99°\text{C}$, $h_g = 2784.2$ kJ/kg

From steam table A.1.3 in the Appendix, using linear interpolation, enthalpy of superheated steam at exit of boiler (12 bar and 274.5°C) is found to be

$h_{\text{sup}} = 2989.29$ kJ/kg

From steam table A.1.1 in the Appendix, enthalpy of water at 45°C,

$h_{f1} = 188.42$ kJ/kg

1. Equivalent evaporation:
2. Boiler efficiency:

Example 2.19

During a trial on an oil-fired smoke tube boiler for 1 h, the following data were recorded:

Steam pressure, $p = 14$ bar

Amount of water evaporated, $m_1 =$ 5400 kg

Condition of steam, $x = 0.92$ dryness fraction

Amount of fuel burnt, $m_f = 540$ kg

Calorific value of fuel used, C.V. = 42000 kJ/kg

Temperature of steam leaving the superheater, T_{sup} = 250°C

Temperature of feed water, t_1 = 50°C

Determine the equivalent evaporation from and at 100°C with and without superheater, boiler efficiency, and the percentage of heat utilised in the superheater.

Solution

Given that p = 14 bar, m_1 = 5400 kg, x = 0.92, m_f = 540 kg, C.V. = 42000 kJ/kg, T_{sup} = 250°C, and t_1 = 50°C

Amount of steam generated per kg of fuel,

Enthalpy of wet steam at 15 bar, 0.92 dry,

$$h = h_f + x.h_{fg} = 845.2 + (0.92 \times 1947.2) = 2636.6 \text{ kJ/kg}$$

From steam table A.1.2 in the Appendix, enthalpy of wet steam at 14 bar, 0.92 dry,

$$h = h_f + x.h_{fg} = 830.29 + (0.92 \times 1959.7) = 2633.2 \text{ kJ/kg}$$

From steam table A.1.3 in the Appendix, Enthalpy of superheated steam at 14 bar, 250°C,

$$h_{\text{sup}} = 2927.2 \text{ kJ/kg}$$

Equivalent evaporation without superheated from and at 100°C.

Equivalent evaporation with superheater from and at 100°C.

Boiler efficiency,

Percentage of heat utilized in the superheater,

Example 2.20

Following particulars refer to a steam power plant consisting of a boiler, superheater, and an economiser:

Steam pressure, $p = 20$ bar

Mass of steam generated, $m_1 =$

10000 kg/h

Mass of coal used, $m_f = 1300$ kg/h

Calorific value of coal, C.V. = 29000 kJ/kg

Temperature of feed water entering the economiser, $t_1 = 35°C$

Temperature of feed water leaving the economiser, $t_2 = 105°C$

Temperature of superheated steam leaving the superheater, $t_{sup} = 350°C$

Dryness fraction of steam leaving the boiler, $x = 0.98$

Determine the following:

1. Overall efficiency of the plant
2. Equivalent evaporation from and at 100°C
3. The percentage of heat utilised in boiler, superheater, and economiser

Solution

Given that p = 20 bar, m_1 = 10000 kg/h, m_f = 1300 kg/h, C.V. = 29000 kJ/kg, t_1 = 35°C, t_2 = 105°C, t_{sup} = 350°C, x = 0.98

From steam table A.1.1 in the Appendix, enthalpy of feed water entering the economiser,

$$h_{f1} = 146.66 \text{ kJ/kg}$$

Enthalpy of feed water leaving the economiser at 105°C,

$$h_{f2} = 440.13 \text{ kJ/kg}$$

From steam table A.1.2 in the Appendix, enthalpy of steam

leaving the boiler,

$$h = h_f + x.h_{fg} = 908.77 + 0.98 \times 1890.7 \text{ (at 20 bar, 0.98 dry)}$$
$$= 2761.66 \text{ kJ/kg}$$

From steam table A.1.3 in the Appendix, enthalpy of steam leaving the superheater,

$$h_{sup} = 3137.0 \text{ kJ/kg (at 20 bar, 350°C)}$$

Mass of steam generated per kg of fuel,

1. Overall efficiency of the plant:
2. Equivalent evaporation from and at 100°C:
3. Percentage of heat utilised in the boiler:

   Heat supplied in 1 kg of fuel = 29000 kJ

   Percentage of heat utilised in superheater:

   Percentage of heat utilised in economiser:

Note: Superheater is considered as a

part of the boiler for calculation of equivalent evaporation.

Example 2.21

The following readings were recorded during a boiler trial of 6 h duration:

Mean steam pressure = 12 bar

Mass of steam generated = 40000 kg

Mean dryness fraction of steam = 0.85

Mean feed water temperature = 30°C

Coal used = 4000 kg

Calorific value of coal = 33400 kJ/kg

Calculate the following:

1. Factor of equivalent evaporation
2. Heat rate of boiler in kJ/hr
3. Equivalent evaporation from and at 100°C
4. Efficiency of the boiler

Solution

Given that p = 12 bar, m = 40000 kg, x = 0.85, t_1 = 30°C, m_f = 4000 kg, C.V. = 33400 kJ/kg, and duration of trial, T = 6 h

From steam table A.1.2 in the Appendix at 12 bar h_f = 798.64 kJ/kg, h_{fg} = 1986.2 kJ/kg

∴ Enthalpy of steam, $h = h_f + x.h_{fg}$ = 798.64 + 0.85 × 1986.2 = 2486.91 kJ/kg

From steam table A.1.1 in the Appendix, enthalpy of feed water at 30°C, h_{f1} = 125.77 kJ/kg

1. Factor of equivalent evaporation:
2. Heat rate of boiler:
3. Equivalent evaporation from and at 100°C:
4. Boiler efficiency:

Example 2.22

During a trial for 8 h on a boiler, the following data were obtained:

Pressure of steam leaving the boiler = 14 bar

Condition of steam = 0.97 dryness

fraction

Steam produced = 26700 kg

Temperature of feed water = 50°C

Mass of coal fired = 4260 kg

Calorific value of coal = 28900 kJ/kg

Air fuel ratio = 17

Temperature of flue gases leaving the boiler = 344°C

Boiler house temperature = 21°C

Specific heat of flue gases = 1.1 kJ/kg.K

Determine the following:

1. Boiler efficiency
2. Equivalent evaporation
3. Heat lost to flue gases expressed as percentage of heat supplied

Solution

Given that p = 14 bar, x = 0.97, m = 26700 kg, t_1 = 50°C, m_f = 4260 kg, C.V. = 28900 kJ/kg, air–fuel ratio, = 17, t_f = 344°C, t_0 = 21°C, C_{pg} = 1.1 kJ/kg.K, and duration of boiler trial, T = 8 h

1. From steam table A.1.2 in the Appendix, at 14 bar, h_f = 830.29 kJ/kg and h_{fg} = 1959.7 kJ/kg

   Enthalpy of steam, $h = (h_f + x.h_{fg})$ at 14 bar = 830.29 + 0.97 × 1959.7 = 2731.2 kJ/kg

   From steam table A.1.1 in the Appendix, enthalpy of water at 50°C, h_{f1} = 209.31 kJ/kg

   ∴ Efficiency of boiler,

2. Equivalent evaporation:

3. Fuel burnt/hour, 532.5 kg/h

Heat supplied in fuel = $\dot{m}_f \times$ C.V. = 532.5 × 28900 = 15389250.0 kJ/h

Mass of flue gases formed/hour,

Heat carried away by flue gases,

$= \dot{m}_g \times c_{pg} (t_f - t_0) = 9585 \times 1.1 (344 - 21) = 3405550.5$ kJ/h

Heat loss of flue gases as per cent of heat supplied

Example 2.23

A gas-fired boiler operates at a pressure of 100 bar. The feed water temperature is 255°C. Steam is produced with a dryness fraction of 0.9, and in this condition, it enters a superheater. Superheated steam leaves the superheater at a temperature of 450°C. The boiler

generates 1200 tonne of steam per hour with an efficiency of 92%. The gas used has a calorific value of 38 mJ/m^3. Determine the following:

1. The heat transfer per hour in producing wet steam in the boiler
2. The heat transfer per hour in producing superheated steam in the superheater
3. The gas used in m^3/hour

Solution

Given that p = 100 bar, t_1 = 256°C, x = 0.9, t_{sup} = 450°C, m = 1200 tonne = 1200 × 10^3 kg

$$\eta = 0.92,\ C.V. = 38\ MJ/m^3 = 38 \times 10^3\ kJ/m^3$$

1. From table A.1.1 in the Appendix, enthalpy of feed water: h_{f1} = 1109.72 kJ/kg

   From table A.1.2 in the Appendix, enthalpy of steam at 100 bar, 0.9 dry

   $$h = h_f + x.h_{fg} = 1407.53 + 0.9 \times 1317.1 = 2592.92\ kJ/kg$$

   From table A.1.3 in the Appendix, Enthalpy of superheated steam at 100 bar, 450°C

$$h_{sup} = 3240.8 \text{ kJ/kg}$$

∴ Heat transfer/h producing wet steam in boiler

$$= m(h - h_{f1}) = 1200 \times 10^3 (2592.92 - 1109.72)$$
$$= 17.7984 \times 10^8 \text{ kJ/h}$$

2. Heat transfer/hour in producing superheated steam in superheater:

$$= m(h_{sup} - h) = 1200 \times 10^3 (3240.8 - 2592.92)$$
$$= 77.7456 \times 10^7 \text{ kJ/h}$$

3. Let the gas used in m^3/hour be V_f.

$$V_f = 73149.2 \text{ m}^3\text{/h}$$

Summary for Quick Revision

1. A pure substance may be defined as a system which is homogeneous in composition and chemical aggregation and invariable in chemical aggregation.
2. Properties of steam are as follows:
    1. Latent heat of steam, h_{fg} = Quantity of heat required to convert 1 kg of liquid water at its boiling point into dry saturated steam at the same pressure.
    2. Dryness fraction, x =

        $$\text{Quality of steam} = x \times 100$$
        $$\text{Wetness fraction} = 1 - x$$

    3. Enthalpy of wet steam, $h_{wet} = h_f + x\, h_{fg}$
    4. Density of wet steam = kg/m^3

        Density of dry saturated steam,

        Volume of wet steam, $v_{wet} = (1 - x)v_f + xv_g = v_f + xv_{fg}$

    5. Internal energy of wet steam, $u_{wet} = u_f + xu_{fg}$
    6. Entropy of wet steam, $s_{wet} = s_f + xs_{fg}$
3. Steam processes are as follows:
    1. Constant volume process,

        $$v_1 = v_2;\ q = u_2 - u_1$$

2. Constant pressure process,

$$w = p\,(p_2 - p_1)$$

$q = (x_2 - x_1)\, h_{fg}$, for wet steam

$= h_{\text{sup}} - h_g$, for superheated steam.

3. Isothermal process,

In the wet region, $q = h_2 - h_1$ and $w = p_1 (v_2 - v_1)$

4. Hyperbolic process,

$$p_1 v_1 = p_2 v_2$$

5. Isentropic process,

$$s_1 = s_2$$

6. Polytropic process,

$n = 1.13$ for wet steam

$= 1.3$ for superheated steam

$$p_1(v_1)^{\mathrm{n}} = p_2(v_2)_n$$

7. Throttling process,

$$h_1 = h_2$$

$$h_1 = h_{f1} + x_1\, h_{fg1} \text{ for wet steam}$$

$$h_2 = h_{f2} + x_2\, h_{fg2} \text{ for wet steam}$$

$$= h_g + c_{ps}\,(T_{\text{sup}} - T_s) \text{ for superheated steam.}$$

4. Determination of dryness fraction

1. Barrel calorimeter,

It gives approximate results which are always lower than the actual value.

2. Separating calorimeter,

The value of x calculated is always greater than the actual.

3. Throttling calorimeter,

   It is suitable for steam with high dryness fraction.

4. Combined separating and throttling calorimeter.

   It is suitable for considerably wet steam.

Multiple-choice Questions

1. The dryness fraction of 1 kg of steam containing 0.8 kg of dry steam is
   1. 0.2
   2. 0.8
   3. 1.0
   4. 0.4
2. If x_1 and x_2 be the dryness fractions of steam obtained in the separating and throttling calorimeters, respectively, then the dryness fraction is
   1. $x_1 + x_2$
   2. x_1x_2
   3. $x_1 - x_2$
   4.
3. Superheating of steam is done at constant
   1. volume
   2. pressure
   3. temperature
   4. enthalpy
4. With the increase of pressure, the enthalpy of evaporation of water
   1. decreases
   2. increases
   3. remains same
   4. changes randomly
5. Only throttling calorimeter is used for measuring
   1. very low dryness fraction up to 0.7
   2. very high dryness fraction up to 0.98
   3. dryness fraction of only low pressure steam
   4. dryness fraction of only high pressure steam
6. Constant pressure lines in the superheated region of the Mollier

diagram have

    1. a positive slope
    2. a negative slope
    3. zero slope
    4. both positive and negative slope

7. Specific volume of wet steam with dryness fraction x is
    1. xv_f
    2. $v_f + x\, v_{fg}$
    3. $x^2 v_g$
    4. $x^3 v_g$
8. Entropy of wet steam is given by
    1. $s_f + x\, s_{fg}$
    2. $x\, s_g$
    3. $s_g + x\, s_{fg}$
    4. $x\, s_f$
9. The phase change at constant pressure (or constant temperature) from liquid to vapour is referred to as
    1. melting
    2. solidification
    3. sublimation
    4. vapourization
10. The point that connects the saturated liquid line to the saturated vapour line is called the
    1. triple point
    2. critical point
    3. superheated point
    4. saturated point

Review Questions

1. Define a pure substance.
2. Define latent heat of fusion and latent heat of vapourisation.
3. Define dryness fraction and wetness of steam.
4. What is quality of steam?
5. Write expressions for internal energy of steam when it is wet and when it is superheated.
6. Write expressions for entropy of steam for wet and superheated.
7. What is a Mollier diagram?
8. When steam is heated at constant volume and its end condition is still wet, what is heat transfer?
9. In the wet region, constant temperature process is also a

constant pressure process. Say true or false.

10. Hyperbolic process is also an isothermal process in the superheat region. Say true or false.
11. What remains constant during isentropic process?
12. What is throttling of steam?
13. What are the drawbacks of barrel calorimeter?
14. Which calorimeter is used for a very wet steam?

Exercises

2.1 A vessel of 1.35 m^3 capacity is filled with steam at 13 bar absolute and 95% dry. The vessel and its contents cool until the pressure is 2 bar absolute. Calculate the mass of the contents in the vessel and the dryness fraction of steam after cooling. Neglect the volume of water present.

[**Ans.** 9.2 kg, 0.1625]

2.2 Steam at a pressure of 5 bar and temperature 200°C is expanded adiabatically to a pressure of 0.7 bar absolute. Determine the final condition

of steam. For superheated steam, c_{ps} = 2.2 kJ/kg.K.

[**Ans.** 0.93]

2.3 Steam at a pressure of 20 bar absolute and dryness fraction 0.8 is throttled to a pressure of 0.5 bar. Determine the final condition of steam.

[**Ans.** 0.905]

2.4 About 2 kg of wet steam at 10 bar and 90% dry is expanded according to the law pv = constant to a pressure of 1 bar. Determine the final condition of steam and the change in internal energy.

[**Ans.** Superheated to 112°C, 257 kJ]

2.5 While conducting the dryness fraction test with a throttling calorimeter, it was found that the

entering steam was at a pressure of 12 bar and a sample after being reduced to 1 bar in the calorimeter was at a temperature of 120°C. Estimate the dryness fraction of steam assuming c_{ps} = 2.1 kJ/kg.K.

[**Ans.** 0.965]

2.6 In a test with a separating and throttling calorimeter, the following observations were made:

Water separated = 2.04 kg

Steam discharged from throttling calorimeter = 20.6 kg at 150°C

Initial pressure of steam = 12 bar absolute

Final pressure of steam = 12.3 cm of mercury above atmospheric pressure

Barometer = 76 cm of mercury

Calculate the dryness fraction of steam entering the calorimeter.

[**Ans.** 0.905]

2.7 Determine the change in internal energy when 1 kg of steam expands from 10 bar and 300°C to 0.5 bar and 0.9 dry. Take c_{ps} = 2.1 kJ/kg.K.

[**Ans.** –669 kJ/kg]

2.8 One kg of steam at 10 bar exists at 200°C. Calculate the enthalpy, specific volume, density, internal energy, and entropy. Take c_{ps} = 2.1 kJ/kg.K.

[**Ans.** 2818 kJ/kg, 0.203 m^3/kg, 4.93 kg/m^3, 2615 kJ/kg, 6.673 kJ/kg.K]

2.9 A vessel contains one kg of steam which contains one-third liquid and two-third vapour by volume. The temperature of steam is 151.86°C. Calculate the quality, specific volume, and specific enthalpy of the mixture.

[**Ans.** 0.0576, 0.003245 m^3/kg, 650.5 kJ/kg]

2.10 A cylinder fitted with a piston contains 0.5 kg of steam at 4 bar. The initial volume of steam is 0.1 m^3. Heat is transferred to steam at constant pressure until the temperature becomes 300°C. Determine the heat transferred and work done during the process.

[**Ans.** 771 kJ, 91 kJ]

2.11 Steam at 10 bar and 0.9 dry initially occupies 0.35 m^3. It is expanded according to the law $pv^{1.25}$ =

constant until the pressure falls to 2 bar. Determine the mass of steam used in the process, the work done, the change in internal energy, and the heat exchange between steam and surroundings.

[**Ans.** 2 kg, 192 kJ, –896 kJ, –704 kJ]

2.12 One kg of steam at 8.5 bar and 0.95 dry expands adiabatically to a pressure of 1.5 bar. The law of expansion is $pv^{1.2}$ = constant. Determine the final dryness fraction of steam and the change in internal energy during the expansion.

[**Ans.** 0.792, –230 kJ/kg]

2.13 A throttling calorimeter is used to measure the dryness fraction of steam in the steam main where the steam is flowing at a pressure of 6 bar. The steam after passing through the calorimeter

comes out at 100 kPa pressure and 120°C temperature. Calculate the dryness fraction of steam in the main. Assume c_{ps} = 2.09 kJ/kg.K.

[**Ans.** 0.985]

2.14 A separating and throttling calorimeter was used to determine the dryness fraction of steam flowing through a steam main at 900 kPa. The pressure and temperature after throttling were 105 kPa and 115°C, respectively. The mass of steam condensed after throttling was 1.8 kg and mass of water collected in the separating calorimeter was 0.16 kg. Determine the dryness fraction of steam flowing through the steam main. Take c_{ps} = 2.09 kJ/kg.K.

[**Ans.** 0.89]

ANSWERS TO MULTIPLE-CHOICE QUESTIONS

1. b
2. b
3. b
4. a
5. b
6. a
7. b
8. a
9. a
10. a

Chapter 3

Steam Generators

3.1 ❒ INTRODUCTION

A steam generator, popularly known as a boiler, is a closed vessel made of high-quality steel in which steam is generated from water by the application of heat. The water receives heat from the hot gases formed by burning fuel through the heating surface of the boiler. Steam is mainly required for power generation, process heating, and space heating purposes.

3.2 ❒ CLASSIFICATION OF STEAM GENERATORS

Steam generators are classified based on

the following:

1. **Water tube and fire tube:** In a water tube boiler, water or steam flows through the tubes and heat is supplied to the external surface. Babcock and Wilcox and Stirling boilers fall under this category.

   In a fire tube boiler, hot gases flow through the tubes and water circulates outside the tubes. There may be one large tube surrounded by water as in a Cornish boiler, or two large tubes surrounded by water in a big tube of water as in the case of Lancashire boiler. There may be many smaller tubes through which hot gases pass and are surrounded by water as in the case of vertical Cochran or locomotive boiler.

2. **Natural/forced circulation:** Natural circulation of water takes place by natural convection currents produced by application of heat as in the case of Lancashire boiler and Babcock and Wilcox boiler.

   In forced circulation, the fluid is forced 'once through' or controlled with partial recirculation as in the case of LaMont boiler, Velox boiler, and Benson boiler.

3. **Horizontal, vertical, or inclined:** The disposition on the principal axis of the boiler decides whether the boiler is horizontal, vertical, or inclined.
4. **Single/multiple tube(s):** A boiler may have only one fire/ water tube or multiple tubes.
5. **Stationary/mobile:** Boilers are called stationary (land) or mobile (marine and locomotive). Stationary boilers are used for power plant steam generation. Mobile boilers are portable

boilers.

6. **Internally/externally fired:** A boiler is said to be an external combustion boiler when combustion takes place outside the region of boiling water. A boiler is said to be an internal combustion boiler if the furnace region is completely surrounded by water-cooled surface as in the case of Lancashire boiler.
7. **Source of heat:** Boilers may be classified according to the source from which heat is supplied to water for evaporation. For example, coal fired, oil fired, gas fired, electrical energy, or nuclear energy.

3.3 ❐ COMPARISON OF FIRE TUBE AND WATER TUBE BOILERS

Table 3.1 shows the comparison of fire tube and water tube boilers.

Table 3.1 *Comparison of fire tube and water tube boilers*

3.4 ❐ REQUIREMENTS OF A GOOD BOILER

A good boiler should possess the following qualities:

1. It should be able to generate steam at the required pressure and quantity with minimum fuel consumption.

2. Its initial cost, installation, maintenance, and operational cost should be as low as possible.
3. It should occupy a small floor area and should be light in weight.
4. It should be able to meet the fluctuating demands of steam without fluctuations in steam pressure.
5. All parts of the boiler should be easily accessible for cleaning, inspection, and maintenance.
6. It should have minimum joints to avoid leakage of steam.
7. Its erection time should be reasonable with minimum labour.
8. The velocities of water and flue gas should be high for better heat transfer with minimum pressure drop through the system.
9. There should be no deposition of mud and foreign materials inside the surface and soot deposits on the outside surface of heat-transferring parts.
10. The boiler should conform to safety precautions as laid down in the Boiler Regulations.

3.5 ❐ FACTORS AFFECTING BOILER SELECTION

The factors affecting boiler selection are as follows:

1. Power to be generated, that is, steam to be raised per hour
2. Working pressure
3. Initial capital investment
4. Facilities available for erection
5. Availability of floor area
6. Location of power house
7. Availability of fuel and water
8. Probable load factor
9. Operating and maintenance cost
10. Accessibility for inspection, cleaning, and maintenance

3.6 ❐ DESCRIPTION OF BOILERS

3.6.1 Fire Tube Boilers

It is a stationary, fire tube, horizontal straight tube, internally fired, natural circulation boiler. Its normal working pressure range is up to 15 atmospheres and evaporative capacity up to approximately 8000 kg/h. The size varies from 7–9 m in length and 2–3 m in diameter.

The three sectional views of the Lancashire boiler are shown in Fig. 3.1. The gaseous products of combustion formed as a result of burning of fuel on the grate move along the furnace tube or main flue. They are deflected up by the brick wall bridge. After reaching the back of the main flue, they deflect downwards and travel through the

bottom flue. The bottom flues are situated below the water shell. These flue gases heat the lower portion of the shell. After travelling from the back to the front, these gases bifurcate into two separate paths in the side flues. They travel from the front to the back again in the main flue from where they discharge to the chimney and atmosphere. The flues are built of ordinary brick-work and lined with refractory material or bricks.

Figure 3.1 *Lancashire boiler*

Superheater

The superheater used with Lancashire boiler is shown in Fig. 3.2. The arrows show the path of the gases, when the superheater is in commission.

When superheated steam is to be produced, the dampers are closed and opened so that the flue gases pass on to the superheater flue at the back and travel towards the bottom flue. In this case, valve 2 is closed and valves 3 and 4 are opened so that steam from the shell passes to the superheater and after getting superheated, they are passed to the main pipe 1. If the superheated steam is not required, valve 2 is opened and valves 3 and 4 are closed and the damper to the superheater flue is closed and the other damper is opened, thus bypassing the superheater and steam is passed through valve 2 to main pipe 1.

Figure 3.2 *Superheater*

Cornish Boiler

In construction and appearance, it is similar to the Lancashire boiler except that here, instead of two flue tubes, only one is used. Its capacity and working pressure range is low. Its shell is usually 4–8 m in length and 1.25–1.75 m in diameter. Its working pressure is about 11.5 bar and can raise steam up to 1350 kg/h.

Locomotive Boiler

The locomotive boiler shown in Fig. 3.3 is an internally fired horizontal multi-tubular portable fire tube boiler. It is specially designed to cope up with sudden variations in demand of steam even at the cost of efficiency. The pressure range is up to about 21 bar and can raise up to 55–70 kg per square

metre of heating surface per hour.

Cochran Boiler

The Cochran boiler, shown in Fig. 3.4, is a multi-tubular, internal furnace vertical fire tube boiler having several horizontal fire tubes. Its normal size is about 2.75 m shell diameter and 6 m height of the shell. The steaming capacity is about 3500 kg/h.

The flame and hot gases produced as a result of burning fuel on the grate, rise in the combustion furnace which is dome shaped. The unburnt fuel, if any, will be deflected. The gases are also deflected back and pass through the flue to the fire tubes from where they pass out to smoke box and hence to chimney.

The water circulation is shown by arrows. Water flows down by cooler wall of the shell and rise up past the fire tubes by natural circulation due to convection currents.

Figure 3.3 *Locomotive boiler*

Figure 3.4 *Cochran boiler*

3.6.2 Water Tube Boilers

For pressures above 10 bar and capacities in excess of 7000 kg of steam per hour, the water tube boiler is used almost exclusively.

Babcock and Wilcox Boiler

Figure 3.5 shows a stationary type Babcock and Wilcox boiler. It consists of a large number of parallel tubes inclined at an angle which varies from

5° to 15° to the horizontal which connect the uptake header with the downtake header. These are connected to the shell having a substantial quantity of water in it. The uptake header is connected to the shell, through a short tube, whereas a long tube is employed to connect the downtake header with the shell. The coal is fed through the fire hole on to the chain grate stoker. The velocity of the chain is adjusted so as to ensure complete combustion of coal by the time it reaches the other end of the grate. The flue gases first rise up, move down, and rise up again due to the presence of the baffles. The hot water and steam moisture rise up through the uptake header into the boiler shell, where steam separates from water and

collects in the steam space. The cold water flows down into the tubes through the downtake header. Hence, a continuous circulation of water is maintained by the convection currents set up.

Superheater

A set of superheater tubes is provided to superheat steam which enters these tubes from the steam space in the boiler shell. The superheated steam can be taken out to the steam stop valve, through the steam pipe. The lower part of the downtake header has a mud box attached to it to collect the sediments. The damper chain controls the quantity of air flowing through the boiler. Two soot doors are provided for internal

cleaning of boiler.

Figure 3.5 *Babcock and Wilcox boiler*

Figure 3.6 *Stirling boiler*

Stirling Boiler

Figure 3.6 shows the Stirling boiler, which comes in a water tube type in which three steam drums are connected together by the banks of bent water tubes. These tubes stand nearly in a vertical position near the rear end while they slope steeply over the fire at the front end. The baffles deflect the products of combustion from the furnace over the banks of tubes into the mud drum. The steam generated collects in the steam spaces of the three drums. The steam can be raised at a working pressure of 60 bar with the temperature

as high as 450°C. An evaporation capacity of 50,000 kg/h has been achieved with a Stirling boiler.

3.7 ❒ HIGH PRESSURE BOILERS

In all modern power plants, boilers raising steam at pressures greater than 100 bar are universally used. These are called high pressure boilers. They offer the following advantages:

1. The efficiency and the capacity of the plant can be increased as reduced quantity of steam is required for the same power generation if high pressure steam is used.
2. The forced circulation of water through the boiler tubes provides freedom in the arrangement of furnace and water walls in addition to the reduction in the heat exchange area.
3. The tendency of scale formation is reduced due to high velocity of water.
4. The danger of overheating is reduced as all the parts are uniformly heated.
5. The differential expansion is reduced due to uniform temperature and this reduces the possibility of gas and air leakages.

3.7.1 Boiler Circulation

There are four types of boiler circulation

as follows.

1. **Natural circulation:** Water circulates naturally in a boiler when its density in one part of the circuit is less than that in another part at the same level. The boiler drums, tubes, and water wall make up the multi-passage circuit. During steam formation in a simple drum, the water near the sides of the drum gets heated and forms steam bubbles, lowering its density. The water in the drum centre being heavier displaces the lighter water-bubble mix and, in turn, gets heated up. This sets up a constant circulation that releases steam into the upper drum region.

   Similarly, in a water tube circuit, the water from the drum flows through the downcomer tube to the bottom of the heated riser. Heat forms steam bubbles in water passing through the riser, lowering the density of the mixture. The heavier downcomer water displaces the riser mixer to establish continuous flow. The force of gravity to produce flow in a natural circulation comes from the difference between the densities of the fluids in the downcomer and riser portions of the circuit. The natural circulation is depicted in Fig. 3.7(a).

2. **Forced circulation:** Forced circulation in various circuits of boiler units is produced by mechanical pumps as depicted in Fig. 3.7(b).
3. **Once-through forced circulation:** This type of circulation, as depicted in Fig. 3.7(c), receives water from the feed supply, pumping it to the inlet of heat-absorbing circuit. Fluid heating and steam generation take place along the length of the circuit until evaporation is completed. Further process through the heated circuits results in superheating the vapour.
4. **Once-through with recirculation (forced):** The

'recirculating' forced-circulating type unit has water supplied to the heat absorbing circuits through a separate circulating pump. The water pumped is considerably in excess of steam produced and requires a steam-and-water drum for steam generation. This type of system is depicted in Fig. 3.7(d).

3.7.2 Advantages of Forced Circulation Boilers

The advantages of forced circulation boilers are as follows:

1. Smaller bore and therefore, lighter tubes
2. Absence from scaling troubles due to high circulation velocity
3. Lighter for a given output
4. Steam can be raised quickly and load fluctuations met rapidly
5. Uniform heating of all parts eliminates the danger of overheating
6. Greater flexibility in layout of boiler parts
7. Boiler can be operated at desired conditions
8. Boiler can be started rapidly

Figure 3.7 *Types of boiler circulation: (a) natural circulation, (b) forced circulation, (c) once-through, (d) once-through with recirculation (force)*

3.7.3 LaMont Boiler

This is a forced circulation boiler whose arrangement of water circulation and different components are shown in Fig. 3.8. Small diameter tubes are used in the evaporating section which gives

flexibility in placing the heat transfer surfaces. The drum may be placed well away from the furnace. These boilers have been built to generate 45–50 tonnes of superheated steam per hour at a pressure of 150 bar, 500°C.

The feed water from the hot well is supplied to the boiler through the economiser. A pump circulates the water through the evaporator tubes and a part of the vapour is separated in the separator drum. The centrifugal pump delivers the water to the headers at a pressure above the drum pressure which distributes the water through the nozzle into the evaporator. The steam separated in the boiler is further passed through the superheater and then supplied to the

prime mover.

Figure 3.8 *LaMont boiler*

3.7.4 Benson Boiler

The major drawback in the LaMont boiler is the formation and sticking of bubbles in the inner surface of the heating tubes. These attached bubbles reduce the heat flow and steam generation as it offers higher thermal resistance as compared to water film. If the boiler pressure is raised to critical pressure (221.2 bar), the steam and water would have same density, and therefore, the danger of bubble formation can be completely eliminated.

The arrangement of the boiler components is shown in Fig. 3.9. The

feed water is passed through the economiser and the radiant evaporator where a major part of water is converted into steam. The remaining water is evaporated by hot gases in the convective evaporator. The supercritical pressure steam is then passed through the superheater after which it is supplied to the prime mover.

The maximum pressure and temperature of steam obtained from this boiler are 500 bar and 650°C, respectively, and generating capacity of 150 tonnes/h. The major difficulty in the operation of this boiler is the deposition of salt and sediments on the inner surface of water tubes.

3.7.5 Loeffler Boiler

The major difficulty of Benson boiler is avoided in the Loeffler boiler by preventing the flow of water into the boiler tubes. Most of the steam is generated outside from the feed water by using a part of the superheated steam coming out from the boiler. The arrangement of different components and steam circulations is shown in Fig. 3.10.

The pressure feed pump draws the water through the economiser and delivers to the evaporator drum. About 65% of the steam coming out of superheater is passed through the evaporator drum to evaporate the feed water coming from the economiser. This steam is drawn by the steam circulating pump and passed

through the radiant superheater and then convective superheater.

Figure 3.9 *Benson boiler*

Figure 3.10 *Loeffler boiler*

Loeffler boilers are available with generating capacity of 94.5 tonnes/h and operating at 140 bar. This boiler can handle high drum salt concentration without trouble.

3.7.6 Schmidt-Hartmann Boiler

The arrangement of this boiler is shown in Fig. 3.11. In the primary circuit, a feed pump supplies water to drum through the economiser, which, in turn, discharges saturated steam to a convective superheater and then to load. A closed secondary circuit heated by furnace resembles a sectional header

boiler. This section supplies the heated vapour in a coil to evaporate the main feed water. It has a tube evaporating section. These boilers are built for pressure ranging from 35–125 bar.

3.7.7 Velox Boiler

The arrangement of a Velox boiler is shown in Fig. 3.12. The pressurised combustion chamber (furnace) of this boiler uses low excess air and has high heat transfer rates. The air for the furnace is supplied by the air compressor. After heating water and steam, the combustion gases pass through a gas turbine to atmosphere. This gas turbine drives the axial flow compressor that raises incoming combustion air from atmosphere to the

furnace pressure. The other important components are feed pump, economiser, steam separating section, water circulating pump, and convective superheater. This boiler is built for pressure ranging from 70–80 bar and 500°C.

Figure 3.11 *Schmidt-Hartmann boiler*

Figure 3.12 *Velox boiler*

3.7.8 Once-through Boiler

The arrangement of this boiler is shown in Fig. 3.13. It has inclined coils arranged in a spiral. Forty coils are paralleled around the furnace. Steam generated in the headers flows into headers and then to the convective superheater. Other essential components are the feed pump and economiser.

The flow of water and steam within the boiler circuit is called circulation. The insulated downcomers carry water from the steam drum to the header and are located outside the furnace. The risers are located inside the furnace and carry steam to the steam drum from the header. Risers installed all around the four-walls of the furnace carry away the heat from the furnace. A simple downcomer–riser circuit connecting a drum and a header is shown in Fig. 3.14.

Downcomers are fewer in number and bigger in diameter, whereas risers are more in number and smaller in diameter. Downcomers are meant to make the water fall by gravity. Bigger the

diameter, lesser the pressure drop due to friction, since pressure drop is inversely proportional to the tube diameter. The pressure drop is given by

where D = diameter of downcomer, L = length of downcomer, ρ = density of water, v = velocity of water, f = Darcy's friction factor. So, the downcomers are made bigger in diameter.

Risers absorb heat from the furnace. For the same total cross-sectional area, the smaller the diameter, the larger the surface area exposed to hot gas for heat transfer. Therefore, the risers are of smaller diameter, as compared to that of downcomers, and more in number.

Figure 3.13 *Once-through boiler*

Figure 3.14 *Downcomer-riser circuit*

3.9 ❒ STEAM DRUM

The purposes of drum used in boiler are as follows:

1. To store water and steam sufficiently to meet varying load requirements.
2. To aid in circulation.
3. To separate vapour or steam from water-steam mixture, discharged by the risers.
4. To provide enough surface area for liquid vapour disengagement.
5. To maintain a certain desired ppm in the drum by phosphate injection and blow down.

A mixture of water, steam, foam and sludge is discharged into the drum by the risers. Steam must be separated from the mixture before it leaves the drum. Any moisture carried with steam to the superheater tubes contains dissolved salts. In the superheater, water evaporates and the salts remain deposited on the inside surface of the

tubes to form a scale, which is difficult to remove. This scale reduces the rate of heat absorption which ultimately leads to the failure of the superheater tubes by overheating and rupture. The superheater tubes are exposed to the highest steam pressure and temperature on the inside and the maximum gas temperature on the outside. The superheater tubes are made of very costly material and utmost care should be exercised so that no damage is done to them by the excessive moisture in steam. Some of the impurities in steam may be vaporised silica, which may cause turbine blade deposits.

No vapour bubbles should flow along with saturated water from the drum to

the downcomers. This will reduce the density difference and the pressure head for natural circulation. The bubbles tending to flow upward may also impede the flow in the downcomer and thus affect circulation. The drum has to secure moisture-free steam going to superheater and bubble-free water going to the downcomers as shown in Fig. 3.15.

Figure 3.15 *Functions of steam drum*

3.9.1 Mechanism of Separation of Moisture in Drum

The separation of steam from steam-water mixture can be done by the following methods:

1. **Gravity separation:** At low pressures up to 20 bar, gravity separation is used if sufficient disengaging surface is provided, as shown in Fig. 3.16(a). Water separates out by gravity from steam-water mixture, being heavier. But at high pressure, separating force due to gravity is low and it is difficult to achieve adequate separation, and there is considerable moisture

carried with steam.

Figure 3.16 *Steam drum separation: (a) Gravity separator, (b) Baffle and screen, (c) Cyclone and scrubber*

2. **Mechanical separators**
    1. **Baffles and screens:** Baffle plates shown in Fig. 3.16(b) act as primary separators. They change or reverse the steam flow direction, thus assisting gravity separation, and act as impact plates that cause water to drain off. Screens made of wire mesh act as secondary separators where the individual wires attract and intercept the fine droplets. The accumulating water drops then fall by gravity back to the main body of water.
    2. **Cyclone separators:** These separators shown in Fig. 3.16(c) utilises the centrifugal forces for separation of two-phase mixture, which is entered tangentially to direct the water downward and to make the steam flow upward. The steam then goes through the zigzag path in corrugated plates, called the scrubber or dryer, on the way out to help remove the last traces of moisture. Finally, perforated plates or screens under the drum exit provide the final drying action.

The internal details of a controlled circulation boiler are shown in Fig. 3.17.

Figure 3.17 *Drum internals of a controlled circulation boiler*

3.10 ❐ FLUIDISED BED BOILER

Fluidised bed boilers produce steam from fossil and waste fuels by using a technique called fluidised bed

combustion (FBC).

3.10.1 Bubbling Fluidised Bed Boiler (BFBB)

A schematic diagram of BFBB is shown in Fig. 3.18. In this boiler, crushed coal (6–20 mm) is injected into the fluidised bed of limestone just above an air-distribution grid at the bottom of the bed.

The air flows upwards through the grid from the air plenum into the bed, where combustion of coal occurs. The products of combustion leaving the bed contain a large proportion of unburnt carbon particles which are collected in cyclone separator and fed back to the bed. The boiler water tubes are located in the furnace.

Since most of the sulphur in coal is retained in the bed by the bed material used (limestone), the gases can be cooled to a lower temperature before leaving the stack with less formation of acid (H_2SO_4). As a result of low combustion temperatures (800–900°C), inferior grades of coal can be used without slagging problem and there is less formation of NO_x. Cheaper alloy materials can also be used, resulting in economy of construction. Further economies are achieved since no pulveriser is required. The volumetric heat release rates are 10 to 15 times higher and the surface heat transfer rates are 2 to 3 times higher than a conventional boiler. This makes the boiler more compact.

Figure 3.18 *Bubbling fluidised bed boiler*

3.10.2 Advantages of BFBB

1. The unit size and hence capital cost are reduced due to better heat transfer.
2. It can respond rapidly to changes in load demand.
3. Low combustion temperatures (800–950°C) restricts the formation of NO_x pollutants.
4. Fouling and corrosion of tubes is reduced considerably due to low combustion temperatures.
5. Cost of coal to fine grind is reduced as it is not essential to grind the coal very fine.
6. Low grade fuels and high-sulphur coal be used.
7. Fossil and waste fuels can be used.
8. Combustion temperature can be controlled accurately.

3.11 ❐ BOILER MOUNTINGS

Different fittings and devices necessary for the operation and safety of a boiler are called boiler mountings. The various boiler mountings are as follows:

1. Water- level indicator
2. Pressure gauge
3. Steam stop valve
4. Feed check valve
5. Blow-down cock
6. Fusible plug
7. Safety valve: spring loaded, dead weight, lever type
8. High steam and low water safety valve.

3.11.1 Water Level Indicator

The water-level indicator indicates the level of water in the boiler constantly. Every boiler is normally fitted with two water-level indicators at its front end. Figure 3.19 shows a water-level indicator used in low pressure boilers. It consists of three cocks and a glass tube. The steam cock D keeps the glass tube in connection with the steam space and cock E puts the glass tube in connection with the water space in the boiler. The drain cock K is used to drain out the water from the glass tube at intervals to ascertain that the steam and water cocks are clear in operation. The glass is generally protected with a shield.

For the observation of water level in the boiler, the steam and water cocks are

opened and the drain cock is closed. The rectangular passage at the ends of the glass tube contains two balls. If the glass tube is broken, the balls are carried along its passage to the ends of the glass tube and flow of water and steam out of the boiler is prevented.

3.11.2 Pressure Gauge

The pressure gauge is used to indicate the steam pressure of the boiler. The gauge is normally mounted in the front top of the steam drum. The commonly used pressure gauge is the Bourdon type pressure gauge shown in Fig. 3.20. It consists of an elastic metallic tube of elliptical cross-section bent in the form of circular arc. One end of the tube is fixed and connected to the steam of the

boiler and the other end is connected to a sector wheel through a link. The sector remains in mesh with a pinion fixed on a spindle. A pointer is attached to the spindle to read the pressure on a dial gauge.

Figure 3.19 *Water-level indicator*

Figure 3.20 *Pressure gauge*

When high pressure steam enters the elliptical tube, the tube section tends to become circular, which causes the other end of the tube to move outward. The movement of the closed end of the tube is transmitted and magnified by the link and sector. The sector is hinged at a point on the link. The magnitude of the movement is indicated by the pointer on the dial.

3.11.3 Steam Stop Valve

The function of the stop valve is to regulate the flow of steam from the boiler to the prime mover as per requirement and shut off the steam flow when not required.

A commonly used steam stop valve is shown in Figure 3.21. It consists of a main body, valve, valve seat, nut, and spindle, which pass through a gland to prevent leakage of steam. The spindle is rotated by means of a hand wheel to close or open.

3.11.4 Feed Check Valve

The function of the feed check valve is to allow the supply of water to the boiler at high pressure continuously and to

prevent the back flow of water from the boiler when the pump pressure is less than boiler pressure or when the pump fails.

A commonly used feed check valve is shown in Fig. 3.22. It is fitted to the shell slightly below the normal water level of the boiler. The lift of the non-return valve is regulated by the end position of the spindle which is attached to the hand wheel. The spindle can be moved up or down with the help of hand wheel which is screwed to the spindle by a nut. Under normal conditions, the non-return valve is lifted due to the water pressure from the pump and water is fed to the boiler. If the case pump pressure falls below boiler pressure, the

valve is closed automatically.

Figure 3.21 *Steam stop valve*

Figure 3.22 *Feed check valve*

3.11.5 Blow-Down Cock

The function of the blow-down cock is to remove sludge or sediments collected at the bottommost point in the water space in a boiler, while the boiler is steaming. It is also used for complete draining of the boiler for maintenance.

A commonly used type of blow-down cock is shown in Fig. 3.23. It consists of a conical plug fitted accurately into a similar casing. The plug has a rectangular opening which may be brought into the line of the passage of the casing by rotating the plug. This

causes the water to be discharged from the boiler. The discharge of water may be stopped by rotating the plug again. The blow-down cock should be opened when the boiler is in operation for quick forcing out of sediments.

3.11.6 Fusible Plug

The main function of the fusible plug is to put off the fire in the furnace of the boiler when the water level in the boiler falls below an unsafe level and thus avoid the explosion which may occur due to overheating of the tubes and shell. The plug is generally fitted over the crown of the furnace or over the combustion chamber.

A commonly used fusible plug is shown in Fig. 3.24. It consists of a hollow gun

metal body screwed into the fire box crown. The body has a hexagonal flange to tighten it into the shell. A gun metal plug is screwed into the gun metal body by tightening the hexagonal flange formed into it. There is yet another solid plug made of copper with conical top and rounded bottom. The fusible metal holds this conical copper plug and the gun metal plug together due to depressions provided at the mating surfaces.

During normal operation, the fusible plug remains submerged in water and on no account, its temperature can be more than the saturation temperature of water.

The fusible metal is protected from direct contact with water or furnace

gases. When the water level in the boiler shell falls below the top of the plug, the fusible metal melts due to overheating. Thus, the copper plug drops down and is held within the gun-metal body by the ribs. The steam space gets communicated to the fire box and extinguishes the fire. The fusible plug can be put into position again by interposing the fusible metal.

Figure 3.23 *Blow-down cock*

Figure 3.24 *Fusible plug*

3.11.7 Safety Valves

The function of a safety valve is to prevent the steam pressure in the boiler exceeding the desired rated pressure by automatically opening and discharging steam to atmosphere till the pressure

falls back to normal rated value. There are three types of safety valves—spring loaded (Ramsbottom) type, dead weight type, and lever type.

1. **Spring-loaded safety valve:** The spring-loaded safety valve of the Ramsbottom type is shown in Fig. 3.25. The spring holds the two valves on their seats by pulling the lever down. The lever is provided with two conical pivots, one integrally forged with the lever and the other pin connected to one end. The upper end of the spring is hooked to the lever midway between the two pivots. The lower end is hooked to the shackle fixed to the valve chest by studs and nuts. The shackle and the lever are also connected by two links, one end of which is pin-jointed and the other end has a slot cut into it to allow for the pin to slide in it vertically, thus allowing the lever to be lifted and retaining the link connection. These links are provided as a safety measure in case the spring breaks, the lever should not fly off. The lever projects on one side for manual operation to check the satisfactory working of the device.
2. **Dead weight safety valve:** In this type of device, as shown in Fig. 3.26, the steam pressure in the upward direction is balanced by the downward force of dead weights acting on the valve. It is generally used on low capacity boilers like the Lancashire boiler. The bottom flange is directly connected to seating block on the boiler shell.
3. **Lever safety valve:** A typical lever safety valve is shown in Fig. 3.27. The lever is the second kind with effort in the middle of fulcrum and load. It is suitable for stationary boilers.

Figure 3.25 *Spring loaded safety valve*

Figure 3.26 Dead weight safety valve

Figure 3.27 *Lever safety valve*

3.11.8 High Steam and Low Water Safety Valve

The functions of water safety valve are of two, namely to blow out if the steam pressure is higher than the working pressure and to blow out steam when the water level in the boiler is low. It is perhaps the most important mounting on the boiler. Figure 3.28 shows the details of Hopkinson's high steam and low water safety valve.

Figure 3.28 *High steam and low water safety valve*

F_1 is the fulcrum, W the weight suspended on one end of the lever, and W is the adjustable balance weight on the other end, which can be fixed in position on the lever 2 by a set screw. A predetermined force is exerted through the strut 1 on the outer valve.

The arrangement of the inner valve with spindle 4 and a dead weight 8 acts as dead weight safety valve. However, the load on the outer valve is both due to dead weight and strut thrust. If the pressure in the boiler exceeds the limit, the valve lifts and the steam escapes to waste.

On one end 9, of lever 6–9 with fulcrum F_2, is attached a float 10 and on the other end is fixed a balance weight 7. When the float 10 is submerged in water, the lever is balanced about fulcrum F_2. As soon as the float is uncovered, it gets unbalanced and tilts the lever to the right. This lever has a projecting knife edge which is clear of the collar 5 when the float is submerged.

As soon as it tilts towards right, it establishes contact with the collar and pushes the spindle 4 up, thus raising the inner valve from its seat. Thus steam starts escaping out through the waste pipe. A drain connection 11 is provided to drain off the condensed steam.

3.12 ❐ BOILER ACCESSORIES

These are the appliances installed to increase the overall efficiency of the steam power plant. The various accessories installed are pressure reducing valve, steam trap, steam separator, economiser, air preheater, superheater, feed pump, and injector.

The functions of the various accessories are as follows:

1. A pressure reducing valve maintains constant pressure on its delivery side with fluctuating boiler pressure.
2. The function of steam trap is to drain of water resulting from the partial condensation of steam from steam pipes and jackets without allowing steam to escape through it.
3. The function of the steam separator is to separate suspended water particles carried by steam on its way from the boiler to the prime mover.
4. The function of an economiser is to recover some of the heat carried away in the flue gases up the chimney and utilise for heating the feed water to the boiler.
5. An air preheater recovers some portion of the waste heat of the flue gases by preheating the air supplied to the combustion chamber.
6. Superheaters are used to increase the temperature of steam above its saturation temperature.
7. The function of a feed pump is to pump water to the water space of the boiler.
8. The function of an injector is to feed water to the boiler with the help of a steam jet.

We shall discuss the air preheater, economiser, and superheater in detail.

3.12.1 Air Preheater

In the air preheater, air is heated by the heat carried away by the flue gases and goes as a waste through the chimney. It is situated between the economiser and the chimney. Air preheaters are of three types—tubular, plate type, and

regenerative.

In the tubular type of air preheater, shown in Fig. 3.29, the air passes down outside the tubes and the flue gases through the tubes before going to the induced draught fan at the base of the chimney. Baffles are provided across the tubes to make the air follow a zigzag path a number of times to utilise more heat of flue gases.

In the plate type of air preheater, there are alternate chambers provided by the plate surfaces for the flow of flue gases upwards to the induced draught fan and for the flow of combustion air down of combustion air downwards.

In the regenerative type, the air

preheater chamber is divided into two rotating compartments. The heating surfaces are made up of a series of corrugated sheets between which the hot flue gases pass upwards on one side and the combustion air passes down on the other side. Due to the rotation of pre-heater at a very low speed, the heater surface comes in contact with combustion air and the cooled surface is again heated by the flue gases. This alternate heating and cooling of heating surface is called regenerative heating and is more effective than other types.

Figure 3.29 *Tubular air preheater*

3.12.2 Economiser

The Green’s economiser shown in Fig. 3.30 is used for boilers of a medium

pressure range up to about 25 bar. It consists of groups of vertical cast iron pipes fitted at their two ends to cast iron boxes, at the top and bottom. All the tubes are enclosed within the brick work of the economiser. There are two pipes, C and D, outside the brick work E. Rows of top and bottom boxes are connected to these top and bottom pipes. The feed water is pumped to the bottom pipe D. From here, it goes to the bottom boxes and rise up in the groups of vertical pipes into the top boxes. From the top boxes, it goes to the pipe C from where it passes to the water space of the boiler through the check valve. F is a stop valve for entry and G is a stop valve for exit of the water from the economiser. S is a safety valve. These

tubes are continuously scraped by scrapers J to remove soot collected on them due to the passage of flue gases through the economiser. A pair of scrapers is connected by a chain passing over a pulley such that one scraper comes down and the other attached to the same chain goes up. L is the soot chamber where soot is collected and removed through a soot door.

Figure 3.30 *Economiser*

3.12.3 Superheater

The superheater commonly used in a Lancashire boiler is shown in Fig. 3.30. It consists of two headers and a set of superheater tubes made of high-quality steel in the form of U-tube. It is located in the path of furnace gases just before

the gases enter the bottom flue. The amount of hot gases passed over the superheater tubes should be in proportion to the amount of superheated steam passing through the tubes. Otherwise, the tubes would be over heated. To avoid this, the hot gases are diverted as shown in Fig. 3.31. The superheater is put out of action by turning the damper upward to the vertical position. In this position of the damper, the gases coming out from the central flue pass directly into the bottom flue without passing over the superheater tubes.

For getting superheated steam, the valves A and B are opened and valve C is closed. The damper is kept open as

per the quantity of steam flowing through the pipe. For this position, the flow direction of steam is shown in Fig. 3.31. If wet steam is required, valves A and B and the gas damper are kept closed, and valve C is kept open. In this case, steam comes directly from the boiler through valve C. By adjusting the gas damper, the temperature of steam coming out of superheater is always maintained constant, irrespective of the amount of steam passing through the superheater.

Figure 3.31 *Superheater*

3.13 ❐ STEAM ACCUMULATORS

Boilers are not always required to supply steam for economic or rated load but also required to meet the fluctuating

demand of load, that is, peak fractional load. The boiler takes an appreciable time to change the rate of steam generation but the demand for a considerable change of power may occur instantly if the boiler could meet such a sudden increase in load; there is always the problem of disposing the steam once the peak has been passed. Moreover, in meeting the fluctuating load, the capacity and efficiency of boiler are reduced. To meet these requirements, it is a general practice to incorporate steam accumulators to equalise the load on the boilers. Thus, the steam accumulators acting as a thermal flywheel store energy during periods when the output of the boiler exceeds the demand and restores or

supplies back the energy when the demand is more than the output of the boiler.

The steam accumulators are of two types, namely variable pressure system (Ruth's accumulator) and constant pressure system (Kiesselbach accumulator).

3.13.1 Variable Pressure Accumulator

This accumulator (Fig. 3.32) was first introduced by Dr Ruth, a Swedish technician, in 1916. It consists of a steel shell filled with about 99% water under low pressure required by the low turbine or process work.

During charging, high pressure steam from the boiler enters the pipe C

through the value V_1. Under this condition, the non-return valve provided in pipe C is closed and the high pressure steam enters into the shell through the non-return valve B. Specially shaped nozzles surrounded by pipes are employed to discharge the steam silently into the water and thereby bring about condensation. This increases the pressure in the accumulator. The more the pressure rises in the accumulator, the greater becomes the thermal storage capacity.

Figure 3.32 *Ruth's accumulator*

During the discharging process (the demand of steam is more than output of boiler), the pressure in the pipe C falls below the pressure of water in the vessel

when the low pressure mains valve is in communication with the industrial plant. In such circumstances, the pressure in the accumulator is also lowered, whereas the temperature remains same as before. This causes evaporation of water into steam, thereby increasing the pressure and reducing temperature until equilibrium condition between temperature and pressure is again attained. The non-return valve B is closed and steam passes to the low pressure turbine or industrial plant through the dome E and the non-return valve A. When the output of the steam equals the consumption of steam, there is no flow into and out of the accumulator.

Ruth's accumulator gives a large discharge over a short period and a small discharge over a large period. This accumulator is widely used in mines and steel works where the reciprocating steam engines direct their intermittent exhausts into a low pressure receiver which is placed in series with accumulators. Low pressure turbines consume the steam supply from the accumulators.

The main advantages of this system are as follows:

1. Increases the efficiency of boiler
2. Increase in rate of evaporation, less repairs, and economy in fuel
3. Rapid pick up
4. Improvement in boiler load factor

3.13.2 Constant Pressure Accumulator

Figure 3.33 shows a constant pressure accumulator. This is used in conjunction with small water tube boilers, it stores heat in water during charging period (i.e., when the demand is less) but delivers water (not steam) during discharge period (i.e., when the demand is more than out-put). Therefore, this is essentially a feed-water accumulator.

In this system, the rate of firing of the boiler is maintained constant but the quantity of feed is varied. When there is fall in the demand for steam, an extra quantity of feed water is circulated through the boiler where the feed-water temperature rises to that of steam, and after that the surplus water is passed into the accumulator.

Figure 3.33 *Kiesselbach accumulator*

3.14 ❐ PERFORMANCE OF STEAM GENERATOR

3.14.1 Evaporation Rate

The quantity of water evaporated into steam per hour is called the evaporation rate. It is expressed as kg of steam/h, kg of steam/h/m^2 of heating surface, kg of steam/h/m^3 of furnace volume, or kg of steam/kg of fuel fired.

Equivalent Evaporation

It is the equivalent of the evaporation of 1 kg of water at 100°C to steam at 100°C. It requires 2257 kJ.

Factor of Evaporation

The ratio of actual heat absorption above feed-water temperature for transformation to steam (wet, dry, or

superheated) to the latent heat of steam at atmospheric pressure (1.01325 bar) is known as a factor of evaporation.

Let h = specific enthalpy of steam actually produced

h_f = specific enthalpy of feed water

$h_{f\,(atm)}$ = specific enthalpy of evaporation at standard atmospheric pressure

m_s = actual evaporation expressed in kg/kg of fuel or kg/h of steam

m_e = equivalent evaporation expressed in kg/kg of fuel or kg/h

Then, the equivalent evaporation = actual evaporation × factor of

evaporation

where F = factor of evaporation

$h_1 = h_{f1} + xh_{fg1}$ for wet steam actually generated

$= h_{g1}$ for saturated steam actually generated

$= h_{sup}$ for superheated steam actually generated.

3.14.2 Performance

The performance of boiler may be explained on the basis of any of the following terms:

1. **Efficiency:** It may be expressed as the ratio of heat output to heat input.
2. **Combustion rate:** It is the rate of burning of fuel in kg/m^3 of grate area/h.

3. **Combustion space:** It is the furnace volume in m^3/kg of fuel fired/h.
4. **Heat absorption:** It is the equivalent evaporation from and at 100°C in kg of steam generated/m^2 of heating surface.
5. **Heat liberated:** It is the heat liberated/m^3 of furnace volume/h.

3.14.3 Boiler Thermal Efficiency

It is the ratio of heat absorbed by steam from the boiler per unit time to the heat liberated by the combustion of fuel in the furnace during the same time.

where $\dot{m}_s$ = mass of steam generated in kg/h

$\dot{m}_f$ = mass of fuel burned in kg/h

C.V. = calorific value of fuel in kJ/kg

3.14.4 Heat Losses in a Boiler Plant

1. **Heat used to generate steam**

$$Q_1 = m_s (h - h_{f1})$$

2. **Heat lost to flue gases:** The flue gases contain dry products of combustion and the steam generated due to the combustion of hydrogen in the fuel.

Heat lost to dry flue gases,

where m_g = mass of gases formed per kg of fuel

c_{pg} = specific heat of gases

T_g = temperature of gases, °C

T_a = temperature of air entering the combustion chamber of the boiler, °C

3. **Heat carried by steam in flue gases**

where m_{s1} = mass of steam formed per kg of fuel due to combustion of H_2 in fuel

h_{f1} = enthalpy of water at boiler house temperature

h_{s1} = enthalpy of steam at the gas temperature and at the partial pressure of the vapour in the flue gas

4. **Heat loss due to incomplete combustion:** If carbon burns to CO instead of CO_2, then it is known as incomplete combustion. 1 kg of C releases 10,200 kJ/kg of heat if it burns to CO, whereas it releases 35,000 kJ/kg if it burns to CO_2. If the percentages of CO and CO_2 in flue gases by volume are known, then

where CO, CO_2 = % by volume of CO and CO_2, respectively, in flue gases

C = fraction of carbon in 1 kg of fuel

Heat lost due to incomplete combustion of carbon per kg of fuel,

5. **Heat lost due to unburnt fuel**

$$Q_5 = m_{f1} \times \text{C.V.}$$

where m_{f1} = mass of unburnt fuel per kg of fuel burnt.

6. **Convection and radiation losses:** Heat unaccounted for due to convection and radiation losses,

where $Q = m_f \times \text{C.V.}$

= heat released per kg of fuel

3.14.5 Boiler Trial and Heat Balance Sheet

There are three purposes of conducting the boiler trial.

1. To determine and check the specified generating capacity of the boiler when working at full load conditions.
2. To determine the thermal efficiency of the plant.
3. To draw up the heat balance sheet so that suitable corrective measures may be taken to improve the efficiency.

The following measurements should be

observed during the boiler trial.

1. The fuel supplied and its analysis.
2. Steam generated and its quality or superheat.
3. Flue gases formed from exhaust analysis.
4. Air inlet temperature and exhaust gases temperature.
5. Volumetric analysis of exhaust gases.
6. Mass of fuel left unburnt in ash.
7. Feed-water temperature.

The heat balance sheet is a systematic representation of heat released from burning of fuel and heat distribution on minute, hour or per kg of fuel basis. A pro forma for heat balance sheet is given in Table 3.2.

Table 3.2 *Heat balance sheet for boiler*

Example 3.1

A boiler generates 4000 kg/h of

steam at 20 bar, 400°C. The feed-water temperature is 50°C. The efficiency of the boiler is 80%. The calorific value of fuel used is 44,500 kJ/kg. The steam generated is supplied to a turbine developing 450 kW and exhausting at 2 bar with dryness fraction 0.96. Calculate the fuel burnt per hour and turbine efficiency. Also find the energy available in exhaust steam above 50°C.

Solution

From steam tables at 20 bar, 400°C, $h_1 = 3245.5$ kJ/kg

Enthalpy of feed water at 50°C, $h_f =$ 209.3 kJ/kg

Fuel used per hour =

Enthalpy of exhaust steam at 2 bar, 0.96 dry,

$$h_2 = h_{f2} + x_2 h_{fg2} = 504.7 + 0.96 \times 2201.9 = 2618.5 \text{ kJ/kg}$$

Enthalpy drop in turbine, $\Delta h = h_1 - h_2 = 3245.5 - 2618.5 = 627.0$ kJ/kg

Turbine efficiency,

Energy available in the exhaust steam above 50°C = $h_2 - h_f$ = 2618.5 – 209.3 = 2409.2 kJ/kg

Example 3.2

The following readings are taken during trial on a boiler for 1 h:

Steam generated = 5000 kg

Coal burnt = 650 kg

CV of coal = 31,500 kJ/kg

Dryness fraction of steam entering the superheater = 0.90

Rated boiler pressure = 10 bar

Temperature of steam leaving the superheater = 250°C

Temperature of hot well = 40°C

Calculate: (a) equivalent evaporation per kg of fuel without and with superheater, (b) thermal efficiency of the boiler without and

with superheater, and (c) amount of heat supplied by the superheater per hour.

Solution

Given that m_s = 5000 kg/h, m_f = 650 kg/h, C.V. of fuel = 31,500 kJ/kg, x = 0.90, p = 10 bar

Equivalent evaporation,

where m_s = mass of steam generated per kg of fuel =

1. Without superheater,

   At 10 bar, h_{f1} = 762.8 kJ/kg, h_{fg1} = 2015.3 kJ/kg

   At 40°C, h_f = 167.6 kJ/kg

   With superheater,

At 10 bar, 250°C, h_{sup} = 2942.6 kJ/kg

2. Thermal efficiency,

Without superheater,

With superheater,

3. Heat supplied by the superheater per hour = $m_s [(1 - x)h_{fg1} + h_{sup}]$

$= 5000 [(1 - 0.9) \times 2015.3 + 2942.6] = 15,720,650$ kJ/h

Example 3.3

The following data refer to a boiler trial:

Feed water = 700 kg/h

Feed-water temperature = 25°C

Steam pressure = 15 bar

Steam temperature = 300°C

Coal burnt = 90 kg

CV of coal = 30,500 kJ/kg

Ash and unburnt coal in ash pit = 4 kg/h with C.V. = 2200 kJ/kg

Flue gas formed = 20 kg/kg of coal burnt

Flue gas temperature at chimney = 300°C

Ambient temperature = 30°C

Mean specific heat of flue gases = 1.025 kJ/kgK

Calculate (a) the boiler efficiency, (b) the equivalent evaporation, and

(c) the percentage heat unaccounted for.

Solution

From steam tables, at 15 bar, 300°C, h_{sup} = 3038.9 kJ/kg

At 25°C, h_f = 108.77 kJ/kg

Heat absorbed by 1 kg of feed water to transform into steam at 300°C and 15 bar

$$= h_{sup} - h_f = 3038.9 - 108.77 = 2930.13 \text{ kJ/kg}$$

1. Boiler efficiency,
2. Equivalent evaporation,

Heat carried away by flue gases = $m_g\, c_{pg}\, (T_g - T_a)$

$$= 20 \times 1.025 \times (300 - 30) = 5535 \text{ kJ/kg of coal}$$

3. Heat unaccounted for per kg of fuel

= CV of fuel – heat in steam – heat in flue gases – heat in ash pit

= 30500 – 22790 – 5535 – 97.8 = 2077.2 kJ/kg

Example 3.4

The following observations were made during a boiler trial:

Mass of feed water and its temperature = 640 kg/h, 50°C

Steam pressure = 10 bar

Fuel burnt = 55 kg/h

HCV of fuel = 44,100 kJ/kg

Temperature of flue gases = 300°C

Boiler house temperature = 30°C

Dryness fraction of steam = 0.97

Heating surface of boiler = 18.6 m^2

Specific heat for flue gases = 1.1 kJ/kgK

Specific heat for superheated steam = 2.1 kJ/kgK

Composition of liquid fuel used by mass: C = 85%, H_2 = 13%, and ash = 2%

The flue gas analysis by volume done by using Orsat apparatus:

$$CO_2 = 12.5\%,\ O_2 = 4.5\%,\ \text{and}\ N_2 = 83\%$$

Calculate (a) the equivalent evaporation per kg of fuel per m^2 of heating surface area per hour, (b) the boiler efficiency, and (c) the

draw the heat balance sheet on 1 kg of fuel basis. Take the partial pressure of steam in exhaust gases = 0.07 bar.

Solution

1. Equivalent evaporation,

$$h_1 = h_{f1} + x_1 h_{fg1}$$

   At 10 bar, h_{f1} = 762.6 kJ/kg, h_{fg1} = 2013.6 kJ/kg

   For feed water at 50°C, h_f = 209.3 kJ/kg

2. Boiler efficiency,
3. Heat used to generate steam per kg of fuel

   Air supplied per kg of fuel burnt =

   Dry flue gases formed per kg of fuel burnt = 17.1 + 0.85 = 17.95 kg

   Heat carried by dry flue gases = $m_g \times c_{pg} (T_g - T_a)$

$$= 17.95 \times 1.1\,(300 - 30) = 5331 \text{ kJ/kg of fuel}$$

   Moisture formed per kg of fuel = 0.13 × 9 = 1.17 kg

Heat carried by moisture in the flue gases per kg of fuel

$$= 1.17\ [h_g + c_{ps}\ (T_{\text{sup}} - T_s) - h_f] \text{ at } 0.07 \text{ bar}$$
$$= 1.17\ [2572.6 + 2.1\ (300 - 39) - 125.8]$$
$$= 3504 \text{ kJ}$$

Heat unaccounted for = 44,100 – (29,166 + 5331 + 3504) = 6099 kJ

Heat balance sheet:

Example 3.5

The following data was recorded during a boiler trial for 1 h:

Steam generated = 4500 kg, Steam pressure = 10 bar gauge

Record of throttling calorimeter:

Steam pressure = 30 mm of Hg

above barometer reading.

Steam temperature = 105°C

Barometer reading = 735 mm of Hg

Specific heat of superheated steam = 2.1 kJ/kgK

Feed-water temperature = 58°C

Temperature of steam leaving the superheater = 202°C

Coal fired = 450 kg, HCV of coal = 35,700 kJ/kg

Flue gas temperature = 260°C,
Boiler house temperature = 30°C

Ultimate analysis of dry coal:

C = 82 %, H_2 = 14 %, and ash = 4%

Volumetric analysis of dry flue gases by Orsat apparatus:

CO_2 = 12 %, CO = 1.5 %, O_2 = 7%, N_2 = 79.5 %

Specific heat of dry flue gases = 1.0 kJ/kgK

Partial pressure of steam going with flue gases = 0.07 bar

Moisture content of coal at the time of feeding into the boiler = 4%

1. Calculate the heat absorbed by the superheater per minute.
2. Draw up the heat balance sheet on the basis of 1 kg of wet coal fired.

Solution

Quality of steam generated by the boiler:

Pressure of steam after throttling =

Absolute pressure of steam generated =

Total heat in steam before throttling at 10.98 bar = Total heat in steam after throttling at 1.02 bar

$$h_{f1} + x_1\, h_{fg1} = (h_{f2} + h_{fg2}) + c_{ps}\,(T_{\text{sup2}} - T_{s2})$$

or $780.936 + x_1 \times 2000.7 = 2675.696 + 2.1\,(105 - 100)$

or $x_1 = 0.952$

Heat absorbed in the superheater per minute:

Heat generated per kg of coal fired:

1. H_g = Content of dry coal × HCV
$$= (1 - 0.04) \times 35{,}700 = 34{,}272 \text{ kJ/kg}$$
2. Heat absorbed in boiler and superheater per kg of coal to generate steam
$$= 10\,[2781.6 + 2.1\,(202 - 184) - 4.187 \times 58] = 25{,}765.8 \text{ kJ/kg}$$
3. Air supplied per kg of coal burned

Dry flue gases formed per kg of coal fired

$$= 14 + (1 - 0.04) \times 0.82 = 14.787 \text{ kg/kg of coal}$$
4. Heat carried away by dry flue gases per kg of coal burned
$$= 14.787 \times 1.0\,(260 - 30) = 3401.0 \text{ kJ/kg of coal}$$
5. Moisture carried away by flue gases per kg of coal burned
= Moisture in the fuel + Moisture formed due to combustion of H_2

Heat carried by moisture going with flue gases

$$= 0.299\,[h_g + c_{ps}\,(T_{\text{sup}} - T_s)] \text{ at } 0.07 \text{ bar}$$
$$= 0.299\,[2572.6 + 2.1\,(260 - 39)] = 908 \text{ kJ}$$
6. Part of carbon burnt to CO =

C.V. when C burns to CO_2 = 35,000 kJ/kg, and when burns to CO = 10,250 kJ/kg

Heat lost due to incomplete combustion = 0.0875 (35,000 – 10,250)

$$= 2165.5 \text{ kJ/kg}$$

Heat balance sheet on the basis of 1 kg of

coal burned is as follows:

Example 3.6

A power plant producing 125 MW of electricity has steam condition at boiler outlet as 110 bar, 500°C and condenser pressure is 0.1 bar. The boiler efficiency is 92 % and consumes coal of calorific value 25 MJ/kg. The feed-water temperature at boiler inlet is 150°C. The steam generator has risers in the furnace wall 30 m high and unheated downcomers. The quality at the top of the riser is 8.5 % and minimum exit velocity of mixture leaving the riser and entering the drum is

required to be 2 m/s. The risers have 50 mm outside diameter and 3 mm wall thickness. Neglecting pump work and other losses, estimate (a) the steam generation rate, (b) the fuel burning rate, (c) the evaporation factor, (d) the pressure head available for natural circulation, (e) the circulation ratio, (f) the number of risers required, and (g) the heat absorption rate per unit projected area of the riser.

Solution

Refer Fig. 3.34.

Given that $P = 125$ MW, $p_1 = 110$ bar, $t_1 = 500°C$, $p_2 = 0.1$ bar, $v_e = 2$ m/s

η_b = 92 %, C.V. = 25 MJ/kg, t_{fw} = 150°C, x_{top} = 0.085

From steam tables:

At p_1 = 110 bar, 500°C, h_1 = 3361 kJ/kg, s_1 = 6.540 kJ/kgK

p_2 = 0.1 bar, h_{f2} = 191.83 kJ/kg, h_{fg2} = 2392.8 kJ/kg

s_{f2} = 0.6493 kJ/kgK, s_{fg2} = 7.5009 kJ/kgK

At t_{fw} = 150°C, h_{f5} = 632.2 kJ/kg

$s_1 = s_2$ for isentropic process

$$6.540 = 0.6493 + x_2 \times 7.5009$$

or x_2 = 0.7853

$$h_2 = h_{f2} + x_2 h_{fg2} = 191.83 + 0.7853 \times 2392.8 = 2070.89 \text{ kJ/kg}$$

$$h_3 = h_4 = h_{f2} = 191.83 \text{ kJ/kg}$$

1. Let $\dot{m}_s$ = mass flow rate of steam in kg/s

   Then $\dot{m}_s (h_1 - h_2) = P$

   or $\dot{m}_s (3361 - 2070.89) = 125 \times 103$

   or $\dot{m}_s = 96.89$ kg/s

2. Boiler efficiency,

Figure 3.34 *Temperature-entropy diagram*

   or

   Rate of fuel burned, $\dot{m}_f = 11.495$ kg/s

3. Evaporation factor =
4. Pressure head available for natural circulation,

   $\Delta p = gH (\rho_f - \rho_m)$

   At 110 bar, $v_f = 0.001489$ m^3/kg, $v_f = 0.015987$ m3/kg

   $v_{top} = 0.001489 + 0.085 (0.015987 - 0.001489) = 0.002721$ m^3/kg

   $\Delta p = 30 \times 9.81 (671.6 - 519.53) = 44.754$ kPa or 0.44754 bar

5. Circulation ratio, CR =
6. $d_i = d_0 - 6 = 50 - 6 = 44$ mm

Rate of steam formation in riser,

Number of risers =

7. At 110 bar, h_{fg} = 1255.5 kJ/kg from steam tables.

Heat absorption rate per unit projected area of the riser

Example 3.7

An oil fired boiler uses 52 kg/h of oil having an HCV of 44,900 kJ/kg and composition/kg: C = 0.847, H_2 = 0.13, S = 0.0125, and generates 635 kg of steam/h at 10.5 bar pressure from feed water supplied at 338 K. The flue gases having dry gas analysis/unit volume: O_2 = 0.043, CO_2 = 0.124, N_2 = 0.833, and specific heat 1.005 kJ/kg-K, leave the boiler at 635 K. The pressure

and temperature of steam after throttling in a throttling calorimeter is 1.15 bar and 398 K, respectively. Taking the partial pressure of steam in flue gases as 0.07 bar and the specific heat of superheated steam = 2.1 kJ/kg-K, determine (a) the equivalent evaporation/kg of fuel and (b) boiler efficiency.

Draw up a complete heat balance sheet. Take boiler room temperature = 234 K. [**IES, 2000**]

Solution

Given: $\dot{m}_f = 52$ kg/h, HCV = 44,900 kJ/kg, $\dot{m}_s = 635$ kg/h, $p_1 = 10.5$ bar,

$T_f = 338$ K, $p_2 = 1.15$ bar, $T_{sup} = 398$

K, p_{vs} = 0.07 bar, T_g = 635 K,

T_a = 234 K, c_{ps} = 2.1 kJ/kgK, c_{pg} = 1.005 kJ/kgK,

Fuel: C = 0.847, H2 = 0.13, S = 0.0125 per kg

Flue gas: O_2 = 0.043, CO_2 = 0.124, N_2 = 0.833

Throttling calorimeter:

$$h_{f1} + x_1\, h_{fg1} = h_{f2} + h_{fg2} + c_{ps}\,(T_{\text{sup}} - T_{s2})$$

From steam tables, we have

At p_1 = 10.5 bar, h_{f1} = 772.2 kJ/kg, h_{fg1} = 2007.7 kJ/kg

At p_2 = 1.15 bar, h_{f2} = 434.0 kJ/kg,

$h_{fg_2} = 2247.5$ kJ/kg, $t_{s_2} = 102.05°C$

or $T_{s_2} = 375.05$ K

$$772.2 + x_1 \times 2007.7 = 434.0 + 2247.5 + 2.1\ (398 - 375.02) = 2729.76$$

or

Dryness fraction of steam = 0.975

1. Equivalent evaporation:

$$h_1 = h_{f1} + x_1\, h_{fg1} = 772.2 + 0.975 \times 2007.7 = 2729.7 \text{ kJ/kg}$$

$$\text{At } T_f = 338 \text{ K},\ h_f = c_{pw}\ (T_f - 273) = 4.187\ (338 - 273) = 272.1 \text{ kJ/kg}$$

$h_{fg(\text{atm})}$ = specific enthalpy of evaporation at standard atmospheric pressure of 1.01325 bar

= 2256.9 kJ/kg

Equivalent evaporation,

2. Boiler efficiency,

Heat balance sheet

1. Heat used to generate steam per kg of fuel =
2. Air supplied per kg of fuel burned =

Dry flue gases formed per kg of fuel burned, $m_g = 17.242 + 0.847 = 18.089$ kg

Heat carried away by dry flue gases = $m_g \times c_{pg} (T_g - T_a)$

$$= 18.089 \times 1.005\ (635 - 234) = 7253.7 \text{ kJ/kg of fuel}$$

3. Moisture formed per kg of fuel, $m_w = 0.13 \times 9 = 1.17$ kg

Heat carried away by moisture in flue gases per kg of fuel

$$= m_w\ [h_g + C_{ps}\ (T_{\text{sup}} - T_s) - h_f] \text{ at } p = 0.07 \text{ bar}$$
$$= 1.17\ [2572.5 + 2.1\ (635 - 312) - 163.4]\ [\because T_s = 39°\text{C at } 0.07 \text{ bar}]$$
$$= 3087.4 \text{ kJ/kg of fuel}$$

4. Heat un-accounted for = 44,900 − (30,011 + 7253.7 + 3087.4)

$$= 4547.9 \text{ kJ/kg of fuel}$$

Example 3.8

During a boiler trial, steam was produced at a pressure of 20 bar and superheat temperature of 360°C using feed water at 30°C at the rate

of 90000 kg/h. Coal with 3% moisture having calorific value of 34000 kJ/kg of dry coal was burnt at the rate of 1000 kg/h. The temperature of flue gases at the exit was 200°C, whereas inlet air was at 30°C. Coal contains 80% carbon, 6% hydrogen, and rest ash. Volumetric analysis of dry flue gases is: CO_2 = 12 %, O_2 = 10% and N_2 = 78 %. Enthalpy of steam at 20 bar, 360° C = 3159.3 kJ/kg, enthalpy of water at 30°C = 126 kJ/kg. In the flue gas, the enthalpy of steam at exit condition is 2879.7 kJ/kg. Specific heat of gas = 1.01 kJ/kg.

Draw the heat balance sheet.

[IES, 1998]

Solution

Steam condition: 20 bar, 360°C

Feed water: 30°C, 90,000 kg/h

Moisture in coal = 3 %, CV of coal = 34000 kJ/kg of dry coal, $\dot{m}_f$ = 1000 kg/h

Exhaust temperature of flue gases = 200°C

Inlet temperature of air = 30°C

Coal composition: C = 80 %, H_2 = 6 %, rest is ash.

Volumetric analysis of dry flue

gases:

CO_2 = 12%, O_2 = 10 %, N_2 = 78 %

Enthalpy of steam at 20 bar, 360°C = 3159.3 kJ/kg

Enthalpy of water at 30°C = 126 kJ/kg

Enthalpy of steam in flue gases at exit = 2879.7 kJ/kg

Specific heat of gas = 1.01 kJ/kg

Heat supplied by coal per kg, $Q = (1 - 0.03)$ C.V. $= (1 - 0.03) \times 34{,}000 = 32{,}980$ kJ/kg

Heat used to generate steam per kg

of fuel,

Air supplied per kg of fuel burnt =

Dry flue gases formed per kg of fuel burned, $m_g = 15.76 + 0.80 = 16.56$ kg

Heat carried away by dry flue gases, $Q_2 = m_g \times c_{pg} (T_g - T_a)$

$$= 16.56 \times 1.01 (200 - 30) = 2843.3 \text{ kJ}$$

Moisture formed per kg of fuel = $0.06 \times 9 = 0.54$ kg

Heat carried away by moisture in flue gases, $Q_3 = 0.54 \times 2879.7 = 1555$ kJ

Heat lost to moisture in coal, Q_4 =

$0.03 \times 2879.7 = 86.39$ kJ

$$Q_1 + Q_2 + Q_3 + Q_4 = 28{,}144 + 2843.3 + 1555 + 86.39 = 32628.69 \text{ kJ}$$

Heat unaccounted for, $Q_5 = 32980 - 32628.69 = 351.31$ kJ

Heat balance sheet:

3.15 ❐ STEAM GENERATOR CONTROL

The purpose of steam generator control is to provide the steam flow required by the turbine at design pressure and temperate. The variables that are controlled are (i) fuel firing rate, (ii) air flow rate, (iii) gas flow distribution, (iv) feed-water flow rate, and (v) turbine valve-setting.

The key measurements that describe the plant performance are (i) steam flow rate, (ii) steam pressure, (iii) steam temperature, (iv) primary and secondary air flow rates, (v) fuel firing rate, (vi) feed-water flow rate, (vii) steam drum level, and (viii) electrical power output.

The control system must act on the measurement of these plant parameters so as to maintain plant operation at the desired conditions.

We shall discuss only a few basic control systems related to the following:

1. **Feed-water and drum level control:** Feed water and steam flow are controlled to meet load demand by the turbine. Therefore, the level of water in the steam drum has to be maintained, normally half-full up to the diametral plane.

   A schematic diagram for the three-element feed-water control system is shown in Fig. 3.35. It comprises a drum-level sensor, feed-water flow

sensor, and steam flow sensor.

Figure 3.35 *Schematic diagram for a three-element feed-water control system*

Figure 3.36 *Schematic diagram of a steam pressure control system*

2. **Steam pressure control:** A schematic diagram of steam pressure control system is shown in Fig. 3.36. It maintains the steam pressure by adjusting fuel and combustion air flow to meet the desired pressure. When pressure drops, the flows are increased. A steam pressure sensor acts directly on the fuel flow and air flow controls. A trimming signal from fuel flow and air flow sensors maintains the proper fuel-air ratio.
3. **Steam temperature control:** *For efficient power plant operation, an accurate control of superheat temperature of steam is essential. The principal variables affecting superheat temperature are* (i) furnace temperature, (ii) cleanliness of radiant and pendant superheaters, (iii) temperature of gases entering the convective superheater, (iv) cleanliness of convective superheater, (v) mass flow rate of gases through the convective superheater, (vi) feed-water temperature, and (vii) variation of load on the unit.

A reduction in steam temperature results in loss in plant efficiency. On the other hand, a rise in steam temperature above design value may result in overheating and failure of superheater, reheater tubes, and turbine blades.

The temperature of the saturated steam

leaving the drum corresponds to the boiler pressure and remains constant if the steam pressure controls are working properly. It is the superheater-reheater responses to load changes which need to be corrected. Some of the methods to correct them are as follows:

1. Combined radiant-convective superheaters
2. Desuperheating and attemperation
3. Gas by-pass or damper control
4. Gas recirculation
5. Excess air
6. Tilting burners
7. Burner selection
8. Separately fired superheater

3.16 ❒ ELECTROSTATIC PRECIPITATOR

The basic elements of an electrostatic precipitator are shown in Fig. 3.37. It comprises two sets of electrodes insulated from each other. The first set is composed of rows of electrically grounded vertical parallel plates, called

the collection electrodes, between which the dust–laden gas flows. The second set of electrodes consists of wires, called the discharge or emitting electrodes that are centrally located between each pair of parallel plates. The wires carry a unidirectional negatively charged high voltage current from an external DC source. The applied high voltage generates a unidirectional, non-uniform electrical field whose magnitude is the greatest near the discharge electrodes. When that voltage is high enough, a blue luminous glow, called a corona, is produced around them. Electrical forces in the corona accelerate the free electrons present in the gas so that they ionise the gas molecules, thus forming more electrons and positive gas ions.

The new electrons create again more free electrons and ions, which result in a chain reaction.

Figure 3.37 *Basic elements of an electrostatic precipitator*

The positive ions travel to the negatively charged wire electrodes. The electrons follow the electrical field toward the grounded electrodes, but their velocity decreases as they move away from the corona region around the wire electrodes towards the grounded plates. Gas molecules capture the low velocity electrons and become negative ions. As these ions move to the collecting electrode, they collide with the fly ash particles in the gas stream and give them negative charge. The negatively charged fly ash particles are driven to the

collecting plate. Collected particulate matter is removed from the collecting plates on a regular schedule.

The arrangement of an electrostatic precipitator is shown in Fig. 3.38.

Figure 3.38 *Arrangement of an electrostatic precipitator*

Example 3.9

A boiler generates 8 kg of steam per kg of coal burnt at a pressure of 10 bar, from feed water having a temperature of 65°C. The efficiency of the boiler is 75% and factor of evaporation 1.2. Specific heat of steam at constant pressure is 2.2 kJ/kgK. Calculate (a) the degree of superheat and temperature of steam

generated, (b) the calorific value of coal used in kJ/kg, and (c) the equivalent evaporation in kg of steam per kg of coal.

Solution

Given: $m_s = 8$ kg/kg of coal, $p_s = 10$ bar, $t_{fw} = 65°C$, $\eta_b = 75\%$, $F_e = 1.2$, $c_{ps} = 2.2$ kJ/kgK

1. At $p = 10$ bar, from steam tables:

   $h_g = 2778.1$ kJ/kg, $T_s = 273 + 179 = 452$ K

   Also, $h_{f1} = c_{pw}\, t_{fw} = 4.187 \times 65 = 272.155$ kJ/kg of feed water

   $h_{fg} = 2256.9$ kJ/kg at atmospheric pressure.

   Factor of evaporation,

   or

   or $2708.28 = 2505.945 + 2.2\,(T_{sup} - 452)$

or $T_{sup} = 543.97$ K or $t_{sup} = 270.97°C$

Degree of superheat = $T_{sup} - T_s = 543.97 - 452$
$= 91.97°C$

2. Boiler efficiency,

or 0.75 =

or C.V. = 28,440 kJ/kg of coal

3. Equivalent evaporation,

Example 3.10

The following observations were made during the trial of a gas-fired boiler of a steam power plant:

Generator output = 50,000 kW

Steam conditions = pressure 50 bar temperature 500°C

Feed-water temperature = 80°C

Steam output = 65 kg/s

Percentage composition of fuel (natural gas), by volume CH_4 = 96.5, C_2H_6 = 0.5 (rest incombustibles)

HCV of gas = 38,700 kJ/m^3, at 1.013 bar and 15°C

Gas consumption = 6.5 m^3/s

Calculate the boiler efficiency and the overall thermal efficiency based on the lower calorific value of the fuel.

[IES, 1996]

Solution

Combustion equations:

$$CH_4 + 2O_2 \rightarrow CO_2 + 2H_2O$$
$$C_2H_6 + 2.5O_2 \rightarrow 2CO_2 + 3H_2O$$

1 mole of CH_4 burns to give 2 moles of water

Therefore, 0.965 moles give $2 \times 0.965 \times 18 = 34.74$ kg of H_2O

Similarly 0.005 moles of C_2H_6 burns to give $3 \times 0.005 \times 18 = 0.27$ kg of H_2O

Assuming steam as a perfect gas, 1 mole of gas at 1.013 bar and OC is 22.41 m^3

$\therefore$ 1 mole of gas at 15°C =

Mass of steam produced,

HCV = LCV + heat in steam

HCV = 38,700 – m_s h_{fg} at 15°C

= 38,700 – 1.481 × 2465.9 [h_{fg} at 15°C = 2465.9 kJ/kg]

= 35,043.07 kJ/kg

At 50 bar and 500°C, h_1 = 3433.8 kJ/kg

At 80°C, h_{fg} for feed water = 334.98 kJ/kg

Enthalpy of steam = $h_1 - h_{fg}$ = 3433.8 – 334.9 = 3098.9 kJ/kg

Boiler efficiency =

Overall thermal efficiency =

Example 3.11

In a trial on a boiler fitted with an economiser, the following results were obtained:

The fuel contains 80 % carbon and it does not contain nitrogen.

Calculate (a) the air leakage into the economiser per minute if the fuel used in the boiler is 60 kg/min. (b) the reduction in temperature of the gas due to air leakage if the

atmospheric air temperature is 20°C and the flue gas temperature is 350°C.

For air: specific heat, c_{pa} = 1.005 kJ/kgK and for gas, c_{pg} = 1.1 kJ/kgK.

Percentage of incombustible in fuel = 15%

Solution

Air supply per kg of fuel for the gas analysis entering the economiser,

Air supply per kg of fuel for the gas analysis leaving the economiser,

Air leakage into the economiser per

kg of fuel burned in boiler

$$= m_{ao} - m_{ai} = 24.7 - 23.4 = 1.3 \text{ kg}$$

Air leakage in the economiser per minute = 1.3 × 60 = 78 kg

Mass of flue gas products formed per kg of coal = 23.4 + (1 − 0.15) = 24.25 kg

$\therefore c_{pg}\, m_g\, T_g + c_{pa}\, m_a\, T_a = c_{pg}\, (m_g + m_a)\, T_m$

where T_m = temperature of exhaust gases leaving the economiser

$$1.1 \times 24.25 \times 623 + 1.005 \times 1.3 \times 293 = 1.1\,(24.25 + 1.3) \times T_m$$

or T_m = 604.9 K or 331.9°C

Drop in temperature of exhaust

gases due to air leakage into the economiser

$$= 350 - 331.9 = 18.1°C$$

3.17 ❐ DRAUGHT

In order to maintain the continuous flow of fresh air into the combustion chamber, it is necessary to exhaust the products of combustion from the combustion chamber of a boiler. A pressure difference has to be maintained to accelerate the products of combustion to their final velocity and to overcome the pressure losses in the flow system. This pressure difference so maintained is called "draught".

3.17.1 Classification of Draught

The method of producing draught may

be classified as follows:

1. Natural (or chimney) draught
2. Artificial draught
    1. Steam jet draught
        1. Induced draught
        2. Forced draught
    2. Mechanical draught
        1. Induced fan draught
        2. Forced fan draught
        3. Balanced draught

3.17.2 Natural Draught

Natural draught is obtained by the use of a chimney. A chimney is a vertical tubular structure of brick, masonry, steel, or reinforced concrete, built for the purpose of enclosing a column of hot gases, to produce the draught. The draught produced by the chimney is due to the density difference between the column of hot gases inside the chimney and the cold air outside.

A diagrammatic arrangement of a

chimney of height H in m above the grate is shown in Fig. 3.39.

Pressure at the grate level on the chimney side,

$$p_1 = p_{atm} + \gamma_g h$$

where p_{atm} = atmospheric pressure at the chimney top

Figure 3.39 *Diagrammatic arrangement of natural chimney draught*

γ_g = specific weight of chimney hot gases.

Pressure acting on the grate on the open (atmospheric) side,

$$p_2 = p_{atm} + \gamma_a h$$

where γ_a = specific weight of air outside the chimney.

Net pressure difference causing the flow

through the combustion chamber, static draught,

Static draught can be measured by a water manometer

Let h_w = difference in water level in the two legs of U-tube manometer, mm

Then

Equating the above two equations, we get

$$\Delta p = h_w = (\gamma_a - \gamma_g)H$$

3.17.3 Height and Diameter of Chimney

It can be assumed that the volume of combustion products is equal to the volume of air supplied when both

reduced to the same temperature and pressure conditions.

Let m_a = mass of air supplied per kg of fuel

T_g = average absolute temperature of chimney gases

T_a = absolute temperature of atmospheric air

Density of air at atmospheric conditions,

Density of hot gases at temperature, T_g,

Let Δp be equivalent to H_1 metre height of burnt gases.

Then

∴

The velocity of gases passing through the chimney is given by,

If the pressure loss in the chimney is equivalent to a hot gas column of height h_1, then

where

= 0.825 for brick chimney

= 1.1 for steel chimney

Mass of gas flowing through any cross-section of chimney is given by,

$$m_g = Av\rho_g \text{ kg/s}$$

3.17.4 Condition for Maximum Discharge Through Chimney

Neglecting losses in chimney,

Density of hot gases,

Mass of gases discharged per second, $m_g = Av\rho_g$

where

The value of m_g will be maximum when , as T_a and m_a are fixed.

Maximum discharge,

A certain minimum flue gas temperature is required to produce a given draught with a given height of chimney. Therefore, the temperature of flue gases leaving the chimney in case of natural draught has to be higher than that of flue gases temperature in the case of artificial draught.

Let T_n = absolute temperature of flue gases leaving the chimney to create the draught of h_w mm of water.

T_m = absolute temperature of flue gases leaving the chimney in case of artificial draught.

c_{pg} = mean specific heat of flue gases

Extra heat carried away per kg of flue gases due to higher temperature required in natural draught = c_{pg} $(T_n - T_m)$ as $T_n > T_m$

Draught pressure produced by natural draught system in m of hot gases,

Maximum energy given by H_1 to one kg of air at the cost of extra heat

Efficiency of chimney,

where $T_n = T_g$

Generally $\eta_{ch} < 1\%$

3.17.6 Advantages and Disadvantages of Natural Draught

Advantages of natural draught are as

follows:

1. No power is required to produce draught.
2. The life of chimney is quite long.
3. The chimney does not require much maintenance.

Disadvantages of natural draught are as follows:

1. There is very low efficiency.
2. It requires a tall chimney (30.48 m minimum).
3. Efficiency is dependent on atmospheric temperature.
4. There is no flexibility on draught.

3.17.7 Draught Losses

The causes of the losses in draught are as follows:

1. Frictional resistance offered by the gas passages during flow of gas.
2. Loss near the bends in the gas flow circuit.
3. Frictional head loss in different equipment, for example, grate, economiser, superheater, etc.
4. Heat lost to impact velocity to flue gases.

The total loss is nearly 20% of the total static draught produced by chimney.

3.17.8 Artificial Draught

A draught produced by artificial means which is independent of the atmospheric conditions is called artificial draught. It gives greater flexibility to take the fluctuating loads on the plant. The artificial draught may be of mechanical type or of steam jet type. If the draught is produced by a fan, it is called fan (or mechanical) draught, and if it is produced by a steam jet, it is called steam jet draught. Steam jet draught is used for small installations such as locomotives, whereas mechanical draught is invariably used in central power stations.

Figure 3.40 *Forced draught*

Forced Draught

In a forced draught system, a blower is installed near the base of the boiler and air is forced to pass through the furnace, flues, economiser, air-preheater, and to the stack. It is called forced positive draught system because the pressure of air throughout the system is above atmospheric pressure and air is forced to flow through the system. The arrangement of the system is shown in Fig. 3.40. A stack or a chimney is also used to discharge gases high into the atmosphere for better dispersion of ash particles and pollutants.

Induced Draught

In this system, the blower is located near the base of the chimney. The air is sucked into the system by reducing the

pressure through the system below the atmospheric pressure. The induced draught fan sucks the gases from the furnace and the pressure inside the furnace is reduced below that of the atmospheric pressure, thus inducing the atmospheric air to flow through the furnace. The draught produced is independent of the temperature of hot gases. Therefore, the gases may be discharged as cold as possible after recovering as much heat as possible in the economiser and the pre-heater. The arrangement of the system is shown in Fig. 3.41.

Figure 3.41 *Induced draught*

3.17.9 Comparison of Forced and Induced Draughts

The comparison of forced and induced

draughts in given in Table 3.3.

Table 3.3 *Comparison of forced and induced draughts*

3.17.10 Comparison of Mechanical and Natural Draughts

The comparison of mechanical and natural draughts is given in Table 3.4.

Table 3.4 *Comparison of mechanical and natural draughts*

3.17.11 Balanced Draught

A balanced draught is a combination of forced and induced draughts. The forced draught overcomes the resistance of the fuel bed and therefore sufficient air is supplied to the fuel bed for proper and complete combustion. The induced

draught fan removes the gases from the furnace, maintaining the pressure in the furnace just below atmospheric pressure. This prevents to blow-out of flames and leakage of air inwards, when the furnace doors are opened. The balanced draught is shown in Fig. 3.42.

Figure 3.42 *Balanced draught*

3.17.12 Steam Jet Draught

A steam jet draught of the induced type is shown in Fig. 3.43. A steam nozzle located near the smoke box induces the flow of gases through the tubes, ash pit, grate, and flues. It is generally used for a locomotive boiler.

The steam jet creates pressure below that of the atmosphere in the smoke box.

The advantages of steam jet draught are as follows:

1. Very simple and economical
2. Low-grade fuel can be used
3. Forced type system keeps the fire bars cool and prevents the adhering of clinker to them
4. Maintenance cost is nil
5. Occupies minimum space

However, it cannot be started until high pressure steam is available.

Figure 3.43 *Steam jet draught*

Example 3.12

A boiler is equipped with a chimney of 25 m height. The ambient temperature is 27°C. The temperature of flue gases passing through the chimney is 300°C. If the air flow through the combustion chamber is 20 kg per kg of fuel

burned, then find the following:

1. Theoretical draught in cm of water
2. The velocity of flue gases passing through the chimney if 50% of theoretical draught is lost in friction at grate and passage.

Solution

Given that $H = 25$ m, $T_a = 27 + 273 = 300$ K, $T_g = 300 + 273 = 573$ K, $m_a = 20$ kg/kg fuel

1. Equivalent gas head,
2. Available head, $H_{av} = 20.476 \times 0.5 = 10.238$ m

   Velocity of flue gases,

Example 3.13

Determine the height of the chimney required to produce a draught equivalent to 2 cm of water

if the flue gas temperature is 240°C and ambient temperature is 25°C. The minimum amount of air per kg of fuel is 18 kg.

Solution

Given that $h_w = 2$ cm of water, $T_g = 240 + 273 = 513$ K, $T_a = 25 + 273 = 298$ K, $m_a = 18$ kg/kg fuel

Example 3.14

Find the mass of flue gases flowing through the chimney when the draught produced is equal to 2 cm of water. The temperature of flue gases is 305°C and the ambient

temperature is 27°C. The flue gases formed per kg of fuel burned are 20 kg. The diameter of the chimney is 2 m. Neglect losses.

Solution

Given that $h_w = 2$ cm, $T_g = 305 + 273 = 578$ K, $T_a = 27 + 273 = 300$ K,

$1 + m_a = 20$ kg/kg of fuel.

Example 3.15

The height of a chimney used in a plant to provide natural draught is 15 m. The ambient air temperature is 20°C. (a) Find the draught in mm

of water when the temperature of chimney gases is such that the mass of gases discharged is maximum and (b) if temperature of gases should not exceed 330°C, then find the air supplied per kg of fuel for maximum discharge.

Solution

Given that $H = 15$ m, $T_a = 20 + 273 = 293$ K, $Tg = 330 + 273 = 603$ K

1. $(h_w)_{\max} =$
2. For maximum discharge,

   or $1.029\ m_a = m_a + 1$

   or $m_a = 34.47$ kg/kg of fuel

Example 3.16

A boiler is equipped with a chimney of 30 m height. The ambient temperature is 25°C. The temperature of flue gases passing through chimney is 300°C. If the air flow is 20 kg/kg of fuel burnt, then find (a) draught produced and (b) the velocity of flue gases passing through chimney if 50% of the theoretical draught is lost in friction.

Solution

$H = 30$ m, $T_a = 273 + 25 = 298$ K, $T_g = 273 + 300 = 573$ K, $m_a = 20$ kg, $m_f = 1$ kg

1. Draught produced,
2. Equivalent gas head,

Available head, $H_{av} = 0.5\, H_1 = 12.469$ m

Velocity of flue gases, v =

Example 3.17

A boiler uses 18 kg of air per kg of fuel. Determine the minimum height of chimney required to produce a draught of 25 mm of water. The mean temperature of chimney gases is 315°C and that of outside air 27°C.

Solution

Given: $m_a = 18$ kg, $m_f = 1$ kg, $h_w =$ 25 mm of H2O, $T_g = 273 + 315 =$ 588 K,

Example 3.18

With a chimney of height 45 m, the temperature of flue gases with natural draught was 370°C. The same draught was developed by induced draught fan and the temperature of the flue gases was 150°C. The air flow through the combustion chamber is 25 kg per kg of coal fired. The boiler house temperature is 35°C. Assuming c_p = 1.004 kJ/kg K for the flue gases, determine the efficiency of the chimney.

Solution

Given that $H = 45$ m, $T_{gn} = 273 + 370 = 643$ K, $T_{gi} = 273 + 150 = 423$ K

m_g = 25 kg/kg of coal fired, T_a = 273 + 35 = 308 K, c_{pg} = 1.004 kJ/kgK

Example 3.19

A boiler is provided with a chimney of 24 m height. The ambient temperature is 25°C. The temperature of flue gases passing through the chimney is 300°C. If the air flow through the combustion chamber is 20 kg/kg of fuel burnt, then find (a) the theoretical draught in cm of water and (b) the velocity of flue gases passing through the

chimney if 50% of theoretical draught is lost in friction at grate and passage.

Solution

Given that $H = 24$ m, $T_a = 273 + 25 = 298$ K, $T_g = 273 + 300 = 573$ K

$m_a = 20$ kg/kg of fuel burnt,

1.
2. Equivalent gas head,

Example 3.20

The following data relate to a boiler using induced draught system:

Length of the duct carrying flue gases = 150 m

Mean size of the square duct = 75 cm2

Mean flue gas velocity in the duct = 900 m/min

Mean temperature of gases passing through duct = 227°C

Plenum pressure = 15 cm of water

Atmospheric pressure = 75 cm of *Hg*

Number of 90° bends = 4

Number of 45° bends = 4

Loss of draft in every 90° bend = 0.1 cm of water

Draft available from chimney = 1.5 cm of water

Fuel bed resistance for under fired stoker = 10 cm of water

Fan efficiency = 60%

Motor efficiency driving fan = 92.5%

Characteristic gas constant for gases = 0.294 kJ/kgK

Friction factor, f, corresponding to round section duct = 0.006

Assuming that a 45° bend equal one-half of 90° bend and that the resistance offered by square duct is

20 % greater than a similar round duct, determine the following:

1. Draft to be produced due to friction in square cross-section duct in mm of water
2. Draft due to velocity head of flue gases in mm of water
3. Total draft for the boiler
4. Draft to be produced by induced fan
5. Power required to drive fan

[IES, 2011]

Solution

Given that $L = 150$ m, $f = 0.006$, $A_s = 75$ cm2, $v_g = 900$ m/min, $C = 1.2$, $\Delta p = 15$ cm of water, $t_g = 227°C$, $p_{atm} = 75$ cm of Hg, $\eta_f = 0.6$, $\eta_m = 0.925$, $R_g = 0.294$ kJ/kgK

1. Frictional head loss in duct carrying flue gases,

   Equivalent diameter of round duct,

   Chimney draft produced, $p_c = 15$ mm of water or $15 \times 9.81 = 147.15$ N/m^2

   Atmospheric pressure,

Pressure of hot flue gases, $p_g = p_{atm} - p_c =$ $0.9992 \times 105 - 147.15$

$= 99{,}844.6$ N/m^2

Density of hot flue gases,

Frictional head loss in square duct in terms of mm of water

= 344 mm of water

Plenum pressure drop in duct = 150 mm of water

Draft to be produced due to friction in duct = 344 + 150 = 494 mm of water

2. Draft due to velocity head of flue gases:

4 bends of 90° =

4 bends of 45° = 0.5 × 41.28 = 20.64 m of hot flue gases

Total draft due to velocity head = 41.28 + 20.64 = 61.92 m of hot flue gases

Loss of draft due to 90° bends = 4 × 1 = 4 mm of water

Loss of draft due to 45° bends = 2 mm of water

3. Fuel bed resistance for under fired stoker = 100 mm of water

Total draft for the boiler = 344 + 150 + 42 + 4 + 2 = 542 mm of water

4. Draft to be produced by induced fan, h = 542 + 100 = 642 mm of H_2O
5. Volume flow rate of flue gases through the duct, $\dot{V} = A_s \times v_g$

$$= 75 \times 10^{-4} \times 15 = 0.1125 \text{ m}^3/\text{s}$$

Power required to drive fan =

Example 3.21

In a steam power plant, the steam generator generates steam at the rate of 120 t/h at a pressure of 100 bar and temperature of 500°C. The calorific value of fuel used by steam generator is 41 MJ/kg with an overall efficiency of 85 %. In order to have efficient combustion, 17 kg of air per kg of fuel is used for

which a draught of 25 mm of water gauge is required at the base of stack. The flue gases leave the steam generator at 240°C. The average temperature of gases in the stack may be taken as 200°C and the atmospheric temperature is 30°C. Determine (i) the height of stack required, and (ii) the diameter of stack at its base. Take the following steam properties for solution: h = 3375 kJ/kg, h_f = 632.2 kJ/kg.

[IAS, 2010]

Solution

Given that m_s = 120 t/h, p = 100 bar, t = 500°C, CV = 41 MJ/kg, η_{overall} = 85%, $m = A/F$ = 17 kg/kg of fuel, h_w

= 25 mm, t_g = 240°C, t_{gs} = 200°C, tatm = 30°C, h = 3375 kJ/kg, h_f = 632.2 kJ/kg

1. Density of air,

   Density of flue gases,

$$\Delta p = 10^3 \times g_{hw} \times 10^{-3} = gH\,(\rho_a - \rho_g)$$
$$25 = H(1.165 - 0.652)$$

   Height of stack, H = 48.73 m

2. Velocity of gases passing through the chimney,

$$\dot{m}_s \times h = \dot{m}_f \times \mathrm{CV} \times \eta_0$$

   Mass of air, $\dot{m}_a = 17 \times \dot{m}_f = 54.878$ kg/s

   Mass of flue gases, $m_g = \dot{m}_f + \dot{m}_a = 3.228 + 54.878 = 58.106$ kg/s

Summary for Quick Revision

1. A boiler is a closed vessel in which steam is produced from water by combustion fuel at the desired temperature and pressure. It consists of a fire place and a steam-raising vessel.
2. In a fire tube boiler, hot gases flow through the tubes and water is heated to raise steam outside the tubes. Cochran, Lancashire, Cornish, Locomotive, and Scotch boilers are fire tube boilers.
3. In water tube boilers, water flows through the tubes and hot

gases heat water from the outside to raise steam. Babcock and Wilcox and Stirling boilers are water tune boilers.

4. High pressure boilers work up to 200 bar pressure and raise steam from 30–650 tons/h. Generally no drum is used in these boilers.
5. Features of boilers are summarised as follows:
6. Critical and supercritical boilers work at more than 221.2 bar pressure.
7. LaMont, Benson, Loeffler, Schmidt–Hartmann, and Velox are high pressure boilers.
8. Boiler mountings are different fittings and devices necessary for the operation and safety of a boiler. The various boiler mountings are: Water level indicator, pressure gauge, steam stop valve, feed check valve, blow down cock, fusible plug, and safety valve.
9. The functions of various boiler mountings are as follows:
    1. **Water-level indicator:** To indicate the level of water in the boiler.
    2. **Pressure gauge:** To indicate steam pressure of the boiler.
    3. **Steam stop valve:** To regulate the flow of steam from the boiler to the prime mover as per requirement.
    4. **Feed check valve:** To allow the supply of water to the boiler at high pressure continuously and to prevent the backflow of water from the boiler when the pump pressure is less than the boiler pressure.
    5. **Blow–down cock:** To remove sludge or sediments collected at the bottommost point in the water space of boiler, when the boiler is working.
    6. **Fusible plug:** To put off the fire in the furnace of the boiler when the water level in the boiler falls below an unsafe level.
    7. **Safety valve:** To prevent the steam pressure in the boiler exceeds the desired rated pressure.
10. Boiler accessories are appliances installed to increase the overall efficiency of the steam power plant. The various accessories are as follows:
    1. Pressure-reducing valve, steam trap, steam separator, economiser, air preheater, superheater, feed pump, and injector.
11. The functions of various accessories are as follows:
    1. **Pressure-reducing valve:** To maintain constant pressure on delivery side with fluctuating boiler

pressure.

2. **Steam trap:** To drain-off water resulting from the partial condensation of steam without steam to escape through it.
3. **Steam separator:** To separate suspended water particles carried by steam on its way from the boiler to prime mover.
4. **Economiser:** To recover some heat carried away by flue gases and use for feed-water heating.
5. **Air preheater:** To recover some heat of flue gases by preheating the air supplied to the combustion chamber.
6. **Superheater:** To increase the temperature of steam above its saturation temperature.
7. **Feed pump:** To pump water to the water space of the boiler.
8. **Injector:** To feed water to the boiler with the help of a steam jet.

12. Evaporation rate is the quantity of water evaporated into steam per hour.
13. Equivalent evaporation is the equivalent of 1 kg of water at 100°C to steam at 100°C. It requires approximately 2257 kJ of heat.
14. Factor of evaporation is the ratio of actual heat absorption above feed-water temperature for transformation to steam to the latent heat of steam at atmospheric pressure.

    Equivalent evaporation = actual evaporation × factor of evaporation

15. Boiler performance parameters:
    1. Efficiency =
    2. Combustion rate =
    3. Combustion space =

    Heat absorption is the equivalent evaporation from and at 100°C in kg of steam generated per m^2 of heating surface

    4. Heat liberated =
16. Boiler efficiency

where $\dot{m}_s$ = mass of steam generated in kg/h

$\dot{m}_f$ = mass of fuel burned in kg/h

C.V. = calorific value of fuel, kJ/kg

17. Boiler power =
18. Heat balance sheet for boiler:
    1. Heat supplied by fuel, $Q = m_f \times$ C.V.
    2. Heat used to generate steam, $Q_1 = m_s \times (h - h_{f1})$
    3. Heat carried away by flue gases, $Q_2 = m_g c_{pg} (T_g - T_a)$
    4. Heat carried away by steam in flue gases, $Q_3 = m_{st} \times (h_1 - h_{f1})$
    5. Heat lost due to incomplete combustion,
    6. Heat lost due to unburnt fuel, $Q_5 = m_{f1} \times$ C.V.
    7. Unaccounted for heat, $Q_6 = Q - (Q_1 + Q_2 + Q_3 + Q_4 + Q_5)$
19. Draught is the small pressure difference maintained in the boiler to exhaust the products of combustion form the combustion chamber.
20. The draught may be natural draught produced by a chimney or artificial draught produced by a steam jet or mechanically.
21. Steam jet draught may be induced or forced type.
22. Mechanical draught may be induced, forced and balanced type.
23. In the induced draught, the blower is located near the base of the chimney, whereas, in the forced draught, the blower is installed near the base of the boiler.
24. Natural draught:

Static draught, $\Delta p = h_w = (\gamma_{\mathrm{a}} - \gamma_{\mathrm{g}}) H$

25. Natural draught in terms of height of burnt gases:
26. Velocity of hot gases passing through the chimney,

where

= 0.825 for brick chimney.

= 1.1 for steel chimney.

27. Chimney diameter,
28. Condition for maximum discharge through chimney.

$$(H_1)_{\max} = H$$

Maximum discharge, $(m_g)_{\max}$ =

29. Chimney efficiency,

where $T_n = T_g$ and T_m = absolute temperature of flue leaving the chimney in case of artificial draught.

η_{ch} is generally less than 1%.

Multiple-choice Questions

1. Consider the following statements:

   Blowdown is necessary on boilers, because

      1. The boiler water level is lowered rapidly in case it accidentally rises too high.
      2. The precipitated sediment of sludge is removed while the boiler is in service.
      3. The concentration of suspended solids in the boiler is controlled.

   Of these statements:

   1. 1, 2, and 3 are correct
   2. 1 and 2 are correct
   3. 3 alone is correct
   4. 1 and 3 are correct
2. Once through boiler is named so because
   1. Flue gas passes only in one direction
   2. There is no recirculation of water
   3. Air is sent through the same direction
   4. Steam is sent out only in one direction
3. Economiser is generally placed between
   1. Last superheater/reheater and air-preheater
   2. Air-preheater and chimney
   3. Electrostatic precipitators
   4. induced draft fan and forced draft fan
4. Attempering is

    1. Reduction of steam pressure
    2. Reduction of steam temperature
    3. Discharge of flue gases at certain height
    4. Conditioning of disposable fluids for minimum pollution
5. In fluidised bed combustion velocity of fluid is proportional to (r = radius of the particle) as
    1.
    2.
    3.
    4. r
6. Which one of the following is the steady flow energy equation for a boiler?
    1.
    2. $Q = (h_2 - h_1)$
    3.
    4. $W_s = (h_2 - h_1) + Q$
7. The output of a boiler is normally stated as
    1. Evaporative capacity in tonnes of steam that can be produced from and at 100°C
    2. Weight of steam actually produced at rated pressure in tonnes per hour
    3. Boiler horse power
    4. Weight of steam produced per kg of fuel
8. Which one of the following is the fire-tube boiler?
    1. Babcock and Wilcox boiler
    2. Locomotive boiler
    3. Stirling boiler
    4. Benson boiler
9. Which one of the following factors has the maximum effect on the equivalent evaporation of a boiler?
    1. Steam generator pressure
    2. Feed-water inlet temperature
    3. Heating surface of the boiler
    4. Quality of steam produced
10. Consider the following:

    1. Safety valve
    2. Steam trap
    3. Steam separator
    4. Economiser

Among these, the boiler accessories would include

1. 1, 2, and 3
2. 2, 3, and 4
3. 1 and 4
4. 1, 2, 3, and 4

11. Match List I and List II and select the correct answer using the codes given below the lists:

Codes:

	A	B	C	D
1.	1	2	3	4
2.	2	3	4	1
3.	3	2	1	4
4.	2	4	1	3

12. Consider the following statements:

    1. Boiler mounting are mainly protective devices.
    2. Steam stop valve is an accessory.
    3. Feed-water pump is an accessory.

Of these statements:

1. 1, 2, and 3 are correct
2. 1 and 2 are correct
3. 2 and 3 are correct
4. 1 and 3 are correct

13. Match List I with List II select the correct answer using the codes given below the list:

Codes:

	A	B	C	D
1.	2	4	3	5
2.	1	3	2	5
3.	3	2	1	4
4.	2	3	1	5

14. In forced circulation boilers, about 90% of water is recirculated without evaporation. The circulation ratio is
    1. 0.1
    2. 0.9
    3. 9
    4. 10
15. Given that h is draught in mm of water, H is chimney height in meters, and T_1 is atmospheric temperature in K, the maximum discharge of gases through a chimney is given by
    1. $h = 176\ T_1/H$
    2. $h = H/176.5\ T_1$
    3. $h = 1.765\ H/T_1$
    4. $h = 176.5\ H/T_1$
16. The excess air required for combustion of pulverised coal is of the order of
    1. 100 to 150%
    2. 30 to 60%
    3. 15 to 40%
    4. 5 to 10%
17. Consider the following:

    1. Increasing evaporation rate using convection heat transfer from hot gases.
    2. Increasing evaporation rate using radiation.
    3. Protecting the refractory walls of the furnace.
    4. Increasing water circulation rate.

The main reasons for providing water wall enclosures in high pressure boiler furnaces would include:

1. 2 and 3

2. 1 and 3
3. 1 and 2
4. 1, 2, 3, and 4

18. Consider the following statements:

Expansion joints in steam pipelines are installed to

1. Allow for future expansion of plant.
2. Take stresses away from flanges and fittings.
3. Permit expansion of pipes due to temperature rise.

Of these statements:

1. 1, 2, and 3 are correct
2. 1 and 2 are correct
3. 2 and 3 are correct
4. 1 and 3 are correct

19. In high pressure natural circulation boilers, the flue gases flow through the following boiler accessories:

1. Superheater
2. Air heater
3. Economiser
4. ID fan

The correct sequence of the flow of flue gases through these boiler accessories is:

1. 1, 3, 4, 2
2. 3, 1, 4, 2
3. 3, 1, 2, 4
4. 1, 3, 2, 4

20. Consider the following components:

1. Radiation evaporator
2. Economiser
3. Radiation superheater

4. Convection superheater

In the case of Benson boiler, the correct sequence of the entry of water through these components is:

1. 1, 2, 3, 4
2. 1, 2, 4, 3
3. 2, 1, 3, 4
4. 2, 1, 4, 3

21. Coal fired power plant boilers manufactured in India generally use
    1. Pulverised fuel combustion
    2. Fluidised bed combustion
    3. Circulating fluidised bed combustion
22. Match List–I (name of the boiler) with List–II (special features) and select the correct answer using the code given below the lists:

Codes:

	A	B	C	D
1.	2	5	1	3
2.	2	4	3	1
3.	1	5	2	3
4.	5	4	1	3

23. Which of the following safety devices is used to protect the boiler when the water level falls below a minimum level?
    1. Water-level indicator
    2. Fusible plug
    3. Blow-off cock
    4. Safety valve
24. Which of the following form part (s) of the boiler mountings?

1. Economiser
2. Feed check valve

3. Steam trap
4. Superheater

Select the correct answer using the codes given below:

Codes:

1. 2
2. 1 and 3
3. 2, 3, and 4
4. 1, 2, 3, and 4

25. Which of the following power plants use heat recovery boilers (unfired) for steam generation?

1. Combined cycle power plants
2. All thermal power plants using coal
3. Nuclear power plants
4. Power plants using fluidised bed combusting

Select the correct answer using the codes given below:

1. 1 and 2
2. 3 and 4
3. 1 and 3
4. 2 and 4

26. Benson boiler is one of the high pressure boilers having
    1. One drum
    2. One water drum and one steam drum
    3. Three drums
    4. No drum
27. Forced draught fans of a large steam generator have
    1. Backward curved blades
    2. Forward curved blades
    3. Straight or radial blades
    4. Double curved blades
28. The device used to heat feed-water by utilising the heat of the exhaust flue gases before leaving through the chimney is called

    1. Superheater
    2. Economiser
    3. Air preheater
    4. ID fan
29. Consider the following statements:
    1. Forced circulation is always used in high pressure power boilers.
    2. Soot blowers are used for cleaning tube surfaces at regular intervals.
    3. Electrostatic precipitator is used to remove fly ash from flue gases.

    Which of these statements are correct?

    1. I, II, and III
    2. II and III
    3. I and III
    4. I and II
30. Once-through boilers operate at
    1. Subcritical pressure
    2. Supercritical pressure
    3. Subcritical as well as supercritical pressures
    4. Critical pressure only
31. Match List I (components) with List II (functions) and select the correct answer using the codes given below the lists:

    Codes:

	A	B	C	D
1.	5	1	4	2
2.	1	3	5	4
3.	5	3	4	2
4.	1	2	5	4
32. Which of the following sequences indicates the correct order for flue gas flow in the steam power plant layout?
    1. Superheater, economiser, and air preheater
    2. Economiser, air preheater, and superheater
    3. Air preheater, economiser, and superheater

4. Economiser, superheater, and air preheater

33. Which one of the following statements is not correct?

In a fluidised bed boiler,

1. The combustion temperatures are higher than those in the conventional boilers
2. Inferior grade of coal can be used without slagging problems
3. The formation of NO_x is less than that in the conventional boilers
4. The volumetric heat release rates are higher than those in the conventional boilers

34. Match List I (boilers) with List II (type/description) and select the correct answer using the codes given below the lists:

Codes:

	A	B	C	D
1.	4	2	1	3
2.	1	4	2	3
3.	4	2	3	1
4.	1	4	3	2

35. A device which is used to drain off water from steam pipes without escape of steam is called
    1. Steam separator
    2. Steam trap
    3. Pressure reducing valve
    4. Injector
36. Blowing down of boiler water is the process to
    1. Reduce the boiler pressure
    2. Increase the steam temperature
    3. Control the solids concentration in the boiler water
    4. Control the drum level
37. Consider the following statements regarding fluidised bed combustion boilers:

    1. The combustion temperatures are low, around

900°C.
2. The formation of oxides of nitrogen is low.
3. It removes sulphur from coal during combustion process.
4. It requires high quality of coal as fuel.

Which of these statements are correct?

1. 1, 2, 3 and 4
2. 1, 2 and 3
3. 2, 3 and 4
4. 1 and 4

38. The correct gas flow path in a typical large modern natural circulation boiler is
    1. Combustion chamber – reheater – superheater – economiser – air preheater – ID fan – electrostatic precipitator – stack
    2. Combustion chamber – superheater – reheater – economiser – air preheater – electrostatic precipitator – ID fan – stack
    3. Combustion chamber – reheater – superheater – air preheater – economiser – electrostatic precipitator – ID fan – stack
    4. Combustion chamber – superheater – reheater – economiser – air preheater – ID fan – electrostatic precipitator – stack
39. The draught in locomotive boilers is produced by
    1. Chimney
    2. Centrifugal fan
    3. Steam jet
    4. Locomotion
40. Consider the following statements regarding waste-heat boilers:

1. Waste-heat boilers are placed in the path of exhaust gases.
2. These are fire tube boilers.
3. The greater portion of the heat transfer in such boilers is due to convection.

Which of the statements given are correct?

1. 1, 2, and 3
2. 1 and 2
3. 2 and 3
4. 1 and 3

41. Which of the following statements is not true for a supercritical steam generator?
    1. It has a very small drum compared to a conventional boiler
    2. A supercritical pressure plant has higher efficiency than a subcritical pressure plant
    3. The feed pressure required is very high, almost about 1.2 to 1.4 times the boiler pressure
    4. As it requires absolutely pure feed water, preparation of feed water is more important than in subcritical pressure boiler
42. Which one of the following is the correct path of water flow through various components of boiler of a modern thermal power plant?
    1. Economiser – boiler drum – water walls – boiler drum – superheater – turbine
    2. Economiser – boiler drum – water walls –superheater – turbine
    3. Economiser – water walls – boiler drum – superheater – turbine
    4. Economiser – water walls – superheater – turbine
43. Consider the following:

    1. Superheater
    2. Economiser
    3. Air pre-heater
    4. Condenser

    Which of these improve overall steam power plant efficiency?

    1. Only 1, 2, and 3
    2. Only 2 and 3
    3. Only 1 and 4
    4. 1, 2, 3, and 4
44. Consider the following:

1. Injector
2. Economiser
3. Blow-off clock
4. Steam stop valve

Which of the above is/are *not* boiler mountings?

1. Only 1
2. Only 1 and 2
3. 1, 2, and 3
4. 2 and 4

45. The draught produced by the chimney is called
    1. Natural
    2. Induced
    3. Forced
    4. Balanced
46. The draught produced by a steel chimney as compared to that produced by a brick chimney for the same height is
    1. More
    2. Less
    3. Same
    4. Unpredictable
47. Artificial draught is produced by
    1. Induced fan
    2. Forced fan
    3. Steam jet
    4. Both (a) and (b)
48. The draught in locomotive is produced by
    1. Forced fan
    2. Induced fan
    3. Steam jet
    4. Chimney
49. The artificial draught produces
    1. Less smoke
    2. More draught
    3. Less chimney gas temperature
    4. All of these
50. The chimney efficiency is approximately
    1. 10%
    2. 5%
    3. 2%
    4. 0.25%

51. The pressure at the furnace is minimum is case of
    1. Forced draught
    2. Induced draught
    3. Balanced draught
    4. Natural draught

Explanatory Notes

1. 14. (c)

Review Questions

1. What is a steam generator?
2. How do you classify steam generators?
3. What are the merits of water tube boilers?
4. What are the demerits of fire tube boilers?
5. List the requirements of a good boiler.
6. Enumerate the factors affecting boiler selection.
7. What are the advantages of high pressure boilers?
8. List the advantages of forced circulation boilers.
9. What is a boiler mounting? List the various boiler mountings.
10. What are the functions of boiler accessories?
11. Name the various accessories used on a boiler.
12. Define equivalent evaporation and factor of evaporation.
13. Define boiler efficiency.
14. What are the purposes of carrying out boiler trial?
15. Differentiate between natural and artificial draught.
16. Describe the consumption features of a Lancashire boiler. How can superheated steam be produced with this boiler?
17. Describe the construction features of a Cochran boiler. Discuss its working briefly.
18. When is a water tube boiler exclusively used? Discuss the construction and working of Babcock and Wilcox boiler.
19. With the help of a neat sketch, describe a Loeffler boiler. What is the working pressure of such a boiler?
20. Draw a schematic flow diagram of a modern steam generator. Discuss the installation of connection and radiation superheaters and their exit temperature response with varying steam flow.
21. With the help of a neat sketch, discuss the working principle of

Velox high pressure boiler.

Exercises

3.1 In a boiler trial, 1360 kg of coal was consumed in 24 hours. The mass of water evaporated was 13600 kg and steam pressure was 7.5 bar. The feed-water temperature was 35°C. The calorific value of 1 kg of coal is 30000 kJ/kg. Calculate the boiler efficiency and equivalent evaporation.

[**Ans.** 87.31%, 11.6 kg]

3.2 A boiler generates 8.5 kg of steam per kg of coal burned at a pressure of 13.5 bar from feed water having absolute temperature of 350 K. The boiler efficiency is 70% and factor of evaporation 1.17. Taking specific heat of steam at constant pressure to be 2.1

kJ/kgK, calculate (a) the degree of superheat and temperature of steam generated, (b) the calorific value of coal, and (c) the equivalent evaporation.

[**Ans.** 84°C, 550.4°C, 32070 kJ/kg, 9.94 kg]

3.3 The data obtained during a boiler trial is as follows:

Duration = 1 hour

Steam generated = 35,500 kg

Steam pressure = 12 bar

Steam temperature = 250°C

Temperature of water entering economiser = 17°C

Temperature of water leaving economiser = 77°C

Oil burnt = 3460 kg

CV of oil = 39,500 kJ/kg

Calculate (a) equivalent evaporation, (b) thermal efficiency of plant, and (c) percentage of heat energy of fuel energy utilised by the economiser.

[**Ans.** 13.01 kg, 74.4%, 6.52%]

3.4 The following data relate to a boiler trial:

Coal burnt per hour = 6750 kg

Moisture content in coal = 2%

HCV of coal = 35,490 kJ/kg

Ultimate mass analysis of dry coal: C = 84%, O_2 = 4%, ash = 8%

Dry flue gas analysis by volume: CO_2 = 9.5%, O_2 = 10.48%, N_2 = 80.02%

Temperature of flue gases = 305°C

Room temperature = 32°C

Specific heat of flue gases = 1 kJ/kgK

Specific heat of steam in flue gases = 2 kJ/kgK

Mass of steam generated at 16.5 bar from feed water at 47°C = 60,500 kg/h

Draw up the heat balance sheet per kg of

dry coal. Assume that air contains 23.1% by mass of oxygen.

Ans.

3.5 Calculate the height of chimney required to produce a draught equivalent to 1.7 m of water if the fuel gas temperature is 270°C and ambient temperature is 22°C and minimum amount of air per kg of fuel is 17 kg.

[**Ans.** 33.45 m]

3.6 A boiler is equipped with a chimney of 30 m height. The flue gases passing through the chimney are at temperature of 228°C, whereas the atmospheric temperature is 21°C. If the air flow through the combustion chamber is 18

kg/kg of fuel burnt, find the theoretical draught produced in mm of water and velocity of flue gases if 50% of theoretical draught is lost.

[**Ans.** 1.37 cm of water, 13.45 m/s]

3.7 Calculate the mass of flue gases flowing through the chimney when the draught produced is equal to 1.9 cm of water. Temperature of flue gases is 290°C and ambient temperature is 20°C. The flue gases formed per kg of fuel burnt are 23 kg. Neglect the losses and take the diameter of the chimney as 1.8 m.

[**Ans.** 38 kg/s]

3.8 A chimney is 42 m high and the temperature of hot gases in the chimney is 310°C. The temperature of outside air

is 35°C. The furnace is supplied with 20 kg of air per kg of fuel burnt. Calculate the draught in cm of water and velocity of flue gases.

3.9 In a chimney of height 45 m, temperature of flue gases with natural draught is 365°C. The temperature of waste gases by using artificial draught is 130°C. The temperature of outside air is 35°C. If air supplied is 20 kg per kg of fuel burnt, determine the efficiency of chimney. Take c_{pg} = 1.004 kJ/kgK for flue gases.

3.10 Calculate the thermal efficiency and equivalent evaporation of a boiler for the following data:

Steam pressure = 10.5 bar

Steam temperature = 250°C

Feed-water temperature = 35°C

Water evaporated = 10 kg/kg of coal burnt

Calorific value of coal = 33,400 kJ/kg

3.11 The following data refers to a steam power plant.

Height of Chimney = 35 m

Draught = 18 mm of water gauge

Temperature of flue gases = 370°C

Boiler house temperature = 32°C

Determine the quantity of air used per

kg of fuel burnt in the boiler.

3.12 In a boiler trial, the following observations were made:

Steam pressure = 10 bar, steam generated = 550 kg/h

Fuel used = 66 kg/h, Moisture in fuel = 1.5% by weight

Weight of dry flue gases = 8.5 kg/kg of fuel, LCV of fuel = 34 MJ/kg

Temperature of flue gases = 320°C, Boiler house temperature = 27°C

Feed-water temperature = 50°C, Specific heat of flue gases = 1.1 kJ/kgK

Dryness fraction of steam = 0.95

Draw up the heat balance sheet for the boiler.

3.13 The following data was obtained during a boiler trial:

Feed-water temperature = 45°C, feed water supplied = 4200 kg/h

Steam pressure = 10 bar, dryness fraction of steam = 0.98

Coal fired = 450 kg/h, CV of dry coal = 37,500 kJ/kg

Moisture in coal = 4%, temperature of flue gases = 280°C

Boiler house temperature = 15°C, barometer pressure = 1 bar,

Ultimate analysis of coal:

C = 86 %, H_2 = 4 %, Ash = 5 %, incombustible matter = 5%

Volumetric analysis of dry flue gases:

CO_2 = 10.4 %, CO = 1.2 %, O_2 = 9 %, N_2 = 79.4 %

Draw the heat balance sheet on 1 kg of coal used basis.

ANSWERS TO MULTIPLE-CHOICE QUESTIONS

1. b
2. b
3. a
4. b
5. c
6. b
7. b
8. b

9. a
10. a
11. c
12. c
13. d
14. c
15. c
16. d
17. c
18. c
19. d
20. c
21. a
22. a
23. b
24. a
25. c
26. d
27. a
28. a
29. a
30. d
31. c
32. a
33. d
34. d
35. b
36. c
37. b
38. b
39. c
40. b
41. c
42. c
43. a
44. b
45. a
46. a
47. d
48. c
49. d
50. d
51. c

Chapter 4

Steam Power Cycles

4.1 ❐ INTRODUCTION

In steam power plants and refrigeration cycles, the working fluid changes from liquid to vapour and back to liquid state. This succession of processes is called vapour cycle. In steam power plants, water is the working fluid in the form of steam and vapour. In refrigeration cycles gasses such as Freon, CO_2, and ammonia (aqua-ammonia) are used as working substances.

4.2 ❐ CARNOT VAPOUR CYCLE

The elements of steam power plants are

as follows:

1. Boiler
2. Steam turbine
3. Condenser
4. Feed pump

The schematic diagram of a steam power plant is shown in Fig. 4.1. The *p*-*v* and *T*-*s* diagrams for the Carnot vapour power cycle are shown in Fig. 4.2(a) and (b). Dry and saturated steam at temperature $T_H = T_1$ enters the steam turbine and is expanded isentropically to sink temperature $T_L = T_2$ to do work. The exhaust steam from the turbine is condensed and cooled in the condenser to state 3 at which the working fluid is in a two-phase mixture of water and its vapour. The water and the vapour are then pumped by the boiler feed pump to state 4 in a saturated state to the boiler,

where it is converted into steam.

Consider 1 kg of the working fluid for analysis.

Work done by steam turbine $w_t = w_{1-2}$ $= h_1 - h_2$

Work done on the pump, $w_p = w_{3-4} = h_{f4} - h_3$

Net work, $w_{net} = w_1 - w_p = (h_1 - h_2) - (h_{f4} - h_3)$

$$= (h_1 - h_{f4}) - (h_2 - h_3) = q_s - q_r$$

Heat supplied, $q_s = h_1 - h_{f4}$

and heat rejected, $q_r = h_2 - h_3$

Figure 4.1 *Schematic diagram of steam power plant*

Figure 4.2 *p-v and T-s diagrams for Carnot vapour cycle: (a) p-v diagram,(b) T-s diagram*

Thermal efficiency,

From the second law of thermodynamics, we have

or $q_s = T_H\,(s_1 - s_4)$

Similarly,

or $q_r = T_L\,(s_2 - s_3)$

Since, $s_1 - s_4 = s_2 - s_3$, one can write

1. **Work Ratio (r_w):** It is defined as the ratio of net work done to turbine work done in the cycle.
2. **Specific Steam Consumption (SSC):** It is defined as the flow rate of steam per unit of power developed. It is expressed in kg/kWh.
3. **Heat Rate:** It is defined as the rate of heat supplied to produce unit work output.

4.2.1 Drawbacks of Carnot Cycle

The major drawbacks of Carnot cycle are as follows:

1. It is very difficult to build a pump which can pump isentropically a two-phase mixture of water and its vapour at state point 3 and to deliver saturated water at state point 4.
2. It is very difficult to superheat steam at constant temperature along path 1–1′.
3. The steam leaving the turbine is of very low dryness fraction. It causes erosion and pitting on turbine blades.
4. The volume of water vapours to be handled by the pump is quite large. It requires a very largesized pump, thus resulting in high power consumption.
5. It requires high specific steam consumption, giving low thermal efficiency. Hence, it is not economically viable.

Example 4.1

A steam power plant operates on an ideal Carnot cycle. Dry saturated steam at 20 bar pressure is supplied to a turbine. It expands isentropically to a condenser pressure of 0.075 bar. Assuming that saturated water enters the boiler, calculate (a) the thermal

efficiency, (b) the work ratio, and (c) the specific steam consumption.

Solution

For dry saturated steam at 20 bar, from the steam tables, we have

$h_1 = h_g = 2799.5$ kJ/kg, $s_1 = s_g = 6.3408$ kJ/kg.K

At 0.075 bar condenser pressure, from the steam tables, we have

$h_{f2} = 168.77$ kJ/kg, $h_{fg2} = 2406.0$ kJ/kg,

$s_{f2} = 0.5763$ kJ/kg.K, $s_{fg2} = 7.6751$ kJ/kg.K

Now $s_1 = s_2 = s_{f2} + x_2\, s_{fg2}$ for the isentropic expansion process 1–2. (Fig.4.2)

$$6.3408 = 0.5763 + x_2 \times 7.6751$$

or $x_2 = 0.751$

$\therefore h_2 = h_{f2} + x_2 h_{fg2} = 168.77 + 0.751 \times 2406.0 = 1975.676$ kJ/kg

At 20 bar, we have

$h_{f4} = 908.77$ kJ/kg, $s_4 = s_{f4} = 2.4473$ kJ/kg.K

For the isentropic compression process 3–4, we have

$s_3 = s_4$

or $s_{f3} + x_3\, s_{fg3} = s_4$

Now $s_{f3} = s_{fg3}$ and $s_{fg3} = s_{fg2}$

$0.5763 + x_3 \times 7.6751 = 2.4473$

or $x_3 = 0.244$

$h_{f3} = h_{f2}$ and $h_{fg3} = h_{fg2}$

$h_3 = h_{f3} + x_3\, h_{fg3} = 168.77 + 0.244 \times 2406.0 = 755.834$ kJ/kg

Pump work, $w_p = h_{f4} - h_3 = 908.77 - 755.834 = 152.936$ kJ/kg

Turbine work, $w_t = h_1 - h_2 = 2799.5 - 1975.676 = 823.824$ kJ/kg

Net work, $w_{\text{net}} = w_t - w_p = 823.824 - 152.936 = 670.888$ kJ/kg

Heat supplied, $q_s = h_1 - h_{f4} = 2799.5 - 908.77 = 1890.73$ kJ/kg

1. Thermal efficiency
2. Work ratio,
3. Specific steam consumption,

4.3 ❐ RANKINE CYCLE

The schematic of Rankine cycle is shown in Fig. 4.3(a). It consists of a turbine, boiler, condenser, and pump. The *p*-*v* diagram is shown in Fig. 4.3(b) and the *T*-*s* diagram in Fig. 4.3(c). The delivery of steam from the boiler takes place at state 1 when assumed dry saturated or 1′ when assumed superheated. The steam expands isentropically in the prime mover to state point 2 from point 1 and 2′ from 1′.

After doing work at point 2 or 2′, the

steam is condensed in the condenser to saturated water, represented by point 3 at pressure p_2. This water is compressed isentropically to pressure p_2 by the pump represented by the process 3–M. Thus, the boiler receives water at pressure p_1 in a sub-cooled state, heats it to temperature corresponding to point 4, and further transforms it into steam at constant pressure p_1. M–4–1 is the operation carried out in the boiler for dry saturated steam and M–4–1′ for superheated steam. All processes except the process of mixing of cold water at point M with hot water at point 4 are reversible. However, when we consider the flow of constant mass, this process gets deleted. The cycle begins again at 1 or 1′.

Figure 4.3 *Rankine cycle: (a) Schematic diagram, (b) p-v diagram, (c) T-s diagram*

4.3.1 Analysis of Rankine Cycle

Consider one kg of mass flow.

Let h_1 = enthalpy of steam at point 1

h_2 = enthalpy of steam at point 2

h_{f3} = enthalpy of water at point 3

h_{f4} = enthalpy of water at point 4

h_{fM} = enthalpy of water at point M

Heat added at constant pressure p_1 represented by M–4–1,

$$q_a = h_1 - h_{f\mathrm{M}} = (h_1 - h_{f3}) - (h_{f\mathrm{M}} - h_{f3})$$

$$= \text{area N–M–4–1–R–N on } T\text{-}s$$

diagram

Heat rejected at constant pressure p_2 represented by 2–3,

$$q_r = h_2 - h_{f3}$$

$$= \text{area R–2–3–N–R on } T\text{-}s \text{ diagram}$$

Net work done, $w_{net} = q_a - q_r$

$$= (h_1 - h_{fM}) - (h_2 - h_{f3})$$
$$= \text{area 1–2–3–M–4–1}$$

Pump work, $w_p = h_{fM} - h_{f3}$

$$w_{net} = (h_1 - h_2) - w_p$$

$$q_a = (h_1 - h_{f3}) - w_p$$

Now $w_p = = v_{f3}\,(p_1 - p_3) \times 10^2$ kJ/kg, where p is in bar.

$= v_{f3} (p_1 - p_2) \times 10^2$ kJ/kg

Thermal efficiency,

Neglecting pump work, which is small as compared to other quantities.

Overall heat rate (OHR) = Specific steam consumption × heat supplied per kg of throttle steam

Example 4.2

In an ideal Rankine cycle, saturated steam vapours enter the turbine at 160 bar and saturated liquid exits the condenser at 0.05 bar. The net power output of the cycle is 130

MW. Determine for the cycle (a) the thermal efficiency, (b) the steam mass flow rate, and (c) the specific steam consumption.

Solution

The Rankine cycle is shown in Fig. 4.4.

From steam tables,

For $p_1 = 160$ bar, $h_1 = h_g = 2580.6$ kJ/kg, $s_1 = s_{g1} = 5.2454$ kJ/kg.K

$p_2 = 0.05$ bar, $v_{f3} = 0.001005$ m^3/kg, $v_{g3} = 28.193$ m^3/kg, $h_{f3} = 137.79$ kJ/kg,

$h_{fg3} = 2423.7$ kJ/kg, $s_{f3} = 0.4763$ kJ/

kg.K, $s_{fg3} = 7.9187$ kJ/kg.K

For insentropic process 1–2,

$s_1 = s_2 = s_{f3} + x_2 s_{fg3}$

or $5.2454 = 0.4763 + x_2 \times 7.9187$

or $x_2 = 0.602$

$h_2 = h_{f3} + x_2 h_{fg3} = 137.79 + 0.602 \times 2423.7 = 1596.86$ kJ/kg

$h_3 = h_{f3} = 137.79$ kJ/kg

$h_M = h_3 + v_{f3} (p_1 - p_2) \times 10^2$

$= 137.79 + 0.001005 (160 - 0.05) \times 10^2 = 153.86$ kJ/kg

Turbine work, $w_t = h_1 - h_2 = 2580.6 - 1596.86 = 983.74$ kJ/kg

Pump work, $w_p = h_M - h_3 = 153.86 - 137.79 = 16.07$ kJ/kg

Heat added $q_a = h_1 - h_M = 2580.6 - 153.86 = 2426.74$ kJ/kg

1. Thermal efficiency =
2. Steam mass flow rate,
3. Specific steam consumption,

Figure 4.4 *Ideal Rankine cycle*

Example 4.3

In a steam power plant working on ideal Rankine cycle, the steam turbine receives steam at 10 bar 250°C and discharges at 0.5 bar. Find the thermal efficiency.

Solution

At p_1 = 10 bar, t_s = 179.91°C.

Therefore, steam is superheated.

The T-s diagram is shown in Fig. 4.5.

Figure 4.5 *Rankine cycle*

From steam tables, at 10 bar, 250°C,

h'_1 = 2942.6 kJ/kg, s'_1 = 6.9246 kJ/kg.K

At p_2 = 0.5 bar, v_{f3} = 0.001030 m^3/kg, h_{f3} = 340.47 kJ/kg, h_{fg3} = 2305.4 kJ/kg, s_{f3} = 1.091 kJ/kg.K,

s_{fg3} = 6.5029 kJ/kg.K,

or $6.9246 = 1.091 + x'_2 \times 6.5029$

or $x'_2 = 0.897$

Pump work, $w_p = v_{f3} (p_1 - p_2) \times 10^2$ = 0.001030 (10 – 0.5) × 10^2 = 0.9785 kJ/kg

Net work output

$$= (2942.6 - 2408.4) - 0.9785 = 533.22 \text{ kJ/kg}$$

Heat added

Thermal efficiency

4.3.2 Effect of Boiler and Condenser Pressure

The Rankine cycle comprises internally reversible processes. Thus, the equation for thermal efficiency can be written in terms of average temperatures of heat addition and heat rejection processes.

With reference to Fig. 4.6, we have

Heat addition, q_a = area $1–b–c–M–4–1$

Figure 4.6 *Heat addition and rejection in a Rankine cycle*

Heat rejected, q_r = area $2–b–c–3–2$

$$= (T_{avg})_{out}\,(s_2 - s_3)$$
$$= (T_{avg})_{out}\,(s_1 - s_M)$$

Thus η_{th} increases if T_{out} decreases or T_{in} increases. Therefore, Eq. (4.11) can be used to study the effects on the performance of the changes in boiler and condenser pressures.

1. **Effect of boiler pressure:** Consider two different cases having same condenser pressure but different boiler pressures as shown in Fig. 4.7(a). The average temperature of heat addition is higher for higher pressure cycle 1′-2′-3-M′-1′ than for cycle 1-2-2-3-M-1. Thus, increase of boiler pressure leads to increase in the thermal efficiency of Rankine cycle.
2. **Effect of condenser pressure:** Figure 4.7(b) shows two ideal Rankine cycles with the same boiler pressure but two different condenser pressures. One condenser operates at atmospheric pressure and the other at pressure less than atmospheric. The temperature of heat rejection for the cycle 1–2″–3″–M″–1 is correspondingly lower than that for cycle 1–2–3–M–1. Thus,

the cycle 1−2″−3″−M″−1 has higher thermal efficiency. Therefore, decreasing the condenser pressure increases the thermal efficiency.

Figure 4.7 *Effect of boiler and condenser pressure on Rankine cycle efficiency: (a) Effect of increase of boiler pressure, (b) Effect of decrease of condenser pressure*

4.4 ❐ METHODS OF IMPROVING EFFICIENCY

The efficiency of the Rankine cycle can be improved by the following methods:

1. Reheating
2. Regeneration
3. Combination of reheating and regeneration

4.4.1 Reheat Cycle

Figure 4.8(a) shows the schematic diagram for the reheat cycle. The corresponding *h*-*s* and *T*-*s* diagrams are shown in Fig. 4.8(b) and Fig. 4.8(c), respectively. The steam is extracted at a suitable point and is reheated with the help of flue gases in the boiler furnace. This increases the dryness fraction of

steam passing through the LP turbine.

Heat supplied, $q_s = (h_1 - h_{f6}) + (h_3 - h_2)$

Pump work, $w_p = h_{f6} - h_{f5}$

Turbine work, $w_t = (h_1 - h_2) + (h_3 - h_4)$

Net work output, $w_{net} = w_t - w_p$

Figure 4.8 *Reheat cycle: (a) Schematic diagram, (b) h-s diagram, (c) T-s diagram*

Figure 4.9 *Reheat cycle with pressure drop between stages: (a) T-s diagram, (b) h-s diagram*

If pump work is neglected,

4.4.2 **Effect of Pressure Drop in the Reheater**

The *T-s* and *h-s* diagrams for a reheat cycle with pressure drop is shown in Fig. 4.9(a) and (b), respectively. Steam

enters the prime mover first stage at point 1. Line 1-2 is the isentropic process. However, in actual practice, it is an irreversible process due to internal frictional resistance offered to the flow of steam. Thus, the condition after the irreversible adiabatic expansion is shown by point 2′. The steam at condition 2′ is passed on the reheater. Since steam cannot pass through the reheater without a pressure drop, the exit from the reheater at point 3 account for the reheater pressure drop from p_2 to $p_2 - \Delta p$. The steam then expands irreversibly and adiabatically to point 4′.

For the first stage, turbine efficiency,

And for the second stage,

Example 4.4

A high pressure boiler delivers steam at 90 bar and 480°C. The steam is expanded in the first stage of turbine to 12 bar and withdrawn and passed on the reheater. The expansion now takes place in the second stage of the steam turbine, down to the condenser pressure of 0.07 bar. Calculate (a) the efficiency of the ideal cycle and (b) the work output and efficiency of the reheat cycle operating through the same states neglecting pump work.

Figure 4.10 *Reheat Rankine cycle*

Solution

The cycle on *T-s* diagram is shown

in Fig. 4.10.

At 90 bar, 480°C, from steam tables,

$h_1 = 3336.5$ kJ/kg, $s_1 = 6.5942$ kJ/kg.K

Now, $s_1 = s_2$

Since, s_g at 12 bar is less than 6.5942, the state 2 is in the superheated region. then, we get

$h_2 = 2814.4$ kJ/kg, $t_2 = 201$°C

At 12 bar, 480°C, $h_3 = 3432.8$ kJ/kg, $s_3 = 7.6198$ kJ/kg.K

At 0.07 bar, $s_{f4} = 0.5589$ kJ/kg.K,

$s_{fg4} = 7.7179$ kJ/kg.K

$$s_3 = s_{f4} + x_4 s_{fg4}$$

or $7.6198 = 0.5589 + x_4 \times 7.7179$

or $x_4 = 0.915$

$h_4 = h_{f4} + x_4 h_{fg4} = 163.40 + 0.915 \times 2409.1 = 2367.7$ kJ/kg

$h_{fs} = h_{f4} = 163.4$ kJ/kg

$h_6 = h_5 + w_p$

$w_p = v_{fs}\,(p_1 - p_2) \times 10^2$

$= 0.001007\,(90 - 0.07) \times 10^2 = 9$ kJ/kg

$h_6 = 163.4 + 9 = 172.4$ kJ/kg

1. $q_a = (h_1 - h_6) + (h_3 - h_2) = (3336.5 - 172.4) + (3432.8 - 2814.4) = 3782.5$ kJ/kg

   $$w_{\text{net}} = (h_1 - h_2) + (h_3 - h_4) - w_p$$

   $$= (3336.5 - 2814.4) + (3432.8 - 2367.7) - 9 = 1578.2 \text{ kJ/kg}$$

2. **Neglecting pump work**

   $$q_a = 3782.5 + 9 = 3781.5 \text{ kJ/kg}$$

   $$w_{\text{net}} = 1578.2 + 9 = 1587.2 \text{ kJ/kg}$$

Example 4.5

In Example 4.4, steam enters the reheater at 12 bar, 210°C, and leaves at 11.5 bar, 480°C. The combined steam rate is 3 kg/kWh, the generator efficiency is 94%, and the heat loss through the turbine casing is 1% of the throttle enthalpy. Determine the turbine

thermal efficiency and condition of exhausted steam.

Solution

The processes in the *T*-*s* diagram is shown in Fig. 4.11.

Figure 4.11 *Reheat Rankine cycle with turbine efficiency*

Refer to Fig. 4.11

= 210°C, = 12 bar, t_3 = 480°C, p_3 = 11.5 bar

Now h_1 = 3336.5 kJ/kg, h_2 = 2814.4 kJ/kg

at 12 bar, 210°C from steam tables

= 2839.8 kJ/kg

h_3 at 11.5 bar, 480°C = 3433.4 kJ/kg

Now

For energy balance of turbine,

$h_1 + h_3 = h'_2 + + w_{net} + q_{loss}$

$= h_1 - + h_3 - w_{net} - 0.01 \times h_1$

$$= 3336.5 - 2839.8 + 3433.4 - 1276.6 - 0.01 \times 3336.5 = 2620.1 \text{ kJ/kg}$$

At p_4 = 0.07 bar, = 2620.1 kJ/kg, = 64°C

Since saturation temperature at 0.07 bar is 39.02°C, the exhaust steam is superheated and the degree of superheat is 24.98°C.

Efficiency of turbine upto extraction

point,

Overall efficiency of turbine,

Example 4.6

Steam at 100 bar and 500°C enters a prime mover that has one stage of reheat. The steam is exhausted from the prime mover at 0.07 bar and 85% dry. The net work developed by the prime mover is 1600 kJ/kg of steam. Calculate the thermal efficiency of the prime mover.

Solution

The cycle on *T-s* diagram is shown

in Fig. 4.12.

Figure 4.12 *Reheat Rankine cycle*

Refer to Fig. 4.12.

From steam tables for p_1 = 100 bar, 500°C,

$h_1 = 3373.6$ kJ/kg

At 0.07 bar, $h_{f5} = 163.38$ kJ/kg

$h_4 = h_{f4} + x_4\, h_{fg4}$

$= 163.38 + 0.85 \times 2409.2 = 2211.2$ kJ/kg

$w_{net} = 1600$ kJ/kg

$$w_{net} = (h_1 - h_2) + (h_3 - h_4)$$

or $h_3 - h_2 = w_{net} + h_4 - h_1 = 1600 +$

2211.2 – 3373.6 = 435.6 kJ/hg

Thermal efficiency η_{th} =

4.5 ❐ REGENERATION

In the Rankine cycle and reheat, the condensate which is at a fairly low temperature is pumped to the boiler. Thus, there is an irreversible mixing of the cold condensate with hot boiler water. This results in loss of cycle efficiency. Regeneration is a method to heat the feed water from the hot well of the condenser reversibly by interchange of heat within the system, thus improving the cycle efficiency. The cycle is called regenerative cycle.

Figure 4.13 *Regenerative heating*

In Fig. 4.13, steam is drawn from the boiler at state point 1 and passes through the turbine and goes to the condenser at state point 2. If no heat is transferred from this steam to the surrounding, which include the casing, and the expansion is isentropic. This process is shown by line 1–2′ in the T-s diagram (Fig. 4.14). However, in the regenerative cycle, it is assumed that the condensate after being pumped to the boiler pressure at state point 4 passed through the hollow casing surrounding the turbine rotor to the boiler. Therefore, the steam losses heat to the surrounding water which gets heated. The temperature of steam entering the turbine and temperature of water leaving the turbine are same.

Let T_1 = saturation temperature of steam at the commencement of expansion at state point 1.

T_3 = temperature of saturated water condensate entering the hollow casing of turbine at state point 3. Thus, 1 kg of water is gradually heated along the path 3–4 and 1 kg of steam gradually loses the same amount of heat during expansion process 1–2. Such a heat exchange within the system is called regenerative heating. Therefore, the net heat supplied to the system is in the boiler during process 4–1 and the net heat rejected from the system is in the condenser during process 2–3 (Fig. 4.14).

Heat added, $q_a = T_1 (s_1 - s_4)$

Heat rejected, $q_r = T_2\ (s_2 - s_3)$

Net work done,

$$w_{\text{net}} = q_a - q_r$$

$$= T_1\ (s_1 - s_4) - T_2\ (s_2 - s_3)$$

But $s_1 - s_4 = s_2 - s_3$

$$\therefore w_{\text{net}} = (T_1 - T_2)\ (s_1 - s_4)$$

Figure 4.14 *T-s diagram for regenerative Rankine cycle*

This method of regeneration is not practicable because the steam becomes extremely wet at the later stages of expansion.

4.5.1 Regenerative Cycle with Open Heaters

The regeneration effect can be achieved by bleeding or extracting small quantities of steam at different points during the expansion and exploiting the energy of bled steam rather than whole steam. Thus, that part of the steam which continues to expand and to do work, does not condense excessively. A regenerative cycle with three bleeding points and therefore, three open type heaters, where feed water and steam mix, is shown in Fig. 4.15. The corresponding *T-s* diagram is shown in Fig. 4.16.

Let 1 kg of steam enter the turbine at state point 1 and expand to the state point 2 where m_1 kg of steam is extracted from the turbine and passed on

to the heater 3. The remainder $(1 - m_1)$ kg of steam continues to expand isentropically to state points 3 where m_2 kg of steam is extracted and passed on the heater 2. The remainder $(1 - m_1 - m_2)$ kg of steam further continues to expand isentropically to state point 4 where m_3 kg of steam is bled and passed on to heater 1. Thus, only $(1 - m_1 - m_2 - m_3)$ kgof steam does the remaining isentropic expansion to point 5 before entering the condenser, where it is condensed to point 6. The condensate $(1 - m_1 - m_2 - m_3)$ kg at enthalpy b_6 is mixed in heater 1. m_3 kg of steam bled at state point 4 condenses to state point 7 and $(1 - m_1 - m_2 - m_3)$ kg of water gets heated from b_6 to point 7. $(1 - m_1 - m_2)$ kg of feed water at state point 7 is

pumped to heater 2 at state point by. In the heater $(1 - m_1 - m_2)$ kg of feed water is heated from state point 7 to state point 8 and m_2 kg of bled steam from point 3 condenses to point 8. Thus, heater 2 has $(1 - m_1)$ kg of feed water at point 8 which is pumped to heater 3 raising the enthalpy of pumped feed water to b_8. This feed water in heater 3 gets heated to point 9 by condensation of m_1 kg of steam bled from point 2, leaving 1kg of feed water in heater 3 to be pumped to boiler pressure by raising its enthalpy to b_9. Heating of 1 kg of feed water from b_9 to point 10 is done in the boiler where it is further heated to steam at point 1.

Figure 4.15 *Regenerative feed water heating with open heaters*

Figure 4.16 *T-s diagram for feed heating with open heaters*

For simplified analysis, points 6 and b_6, 7 and b_7, 8, and b_8, 9 and b_9 are treated as coincident points, thus ignoring the very small quantity of pump work.
Thus, we have

Heat gained by feed water = Heat lost by bled steam

Heater 1: $(1 - m_1 - m_2 - m_3)(h_7 - h_6) = m_3 (h_4 - h_7)$

Heater 2: $(1 - m_1 - m_2)(h_8 - h_7) = m_2 (h_3 - h_8)$

Heater 3: $(1 - m_1)(h_9 - h_8) = m_1 (h_2 - h_9)$

Net work output, $w_{out} = (h_1 - h_2) + (1 - m_1)(h_2 - h_3) + (1 - m_1 - m_2)(h_3 - h_4) +$

$(1 - m_1 - m_2 - m_3)(h_4 - h_5)$

$$w_{\text{net}} = w_{\text{out}} - \sum w_p$$

Heat added, $q_a = h_1 - h_{b9} = h_1 - h_9 - w_{p4}$

Now $\sum w_p = (1 - m_1 - m_2 - m_3)(h_{b6} - h_6) + (1 - m_1 - m_2)(h_{b7} - h_7) + (1 - m_1)(h_{b8} - h_8) + (h_{b9} - h_9)$

$= v_{f6}(p_1 - p_6)$

where p_1 = throttle pressure, p_6 = condenser pressure and v_{f6} = specific volume of water at condenser pressure.

4.5.2 Regenerative Cycle with Closed Heaters

In this type of heater, the feed water is contained in the tubes about which the bled steam passes and condenses. One

pump can be used for two or more heaters placed in series. The heaters can be placed in various combinations. One such arrangement is shown in Fig. 4.17. The corresponding *T-s* diagram is shown in Fig. 4.18.

Work done per kg of steam supplied to turbine,

$$w_t = (h_1 - h_2) + (1 - m_1)(h_2 - h_3) + (1 - m_1 - m_2)(h_3 - h_4)$$

Total pump work, $\sum w_p = w_{p1} + w_{p2} + w_{p3}$

$$= (1 - m_1 - m_2)(h_6 - h_5) + m_2 (h_{10} - h_9) + m_1 (h_{12} - h_{11})$$

where $h_6 - h_5 = v_{f5} (p_1 - p_4) \times 10^2$ kJ/kg

$h_{10} - h_9 = v_{f9} (p_1 - p_3) \times 10^2$ kJ/kg

$h_{12} - h_{11} = v_{f11} (p_1 - p_2) \times 10^2$ kJ/kg

where v_f is in m^3/kg and p is in bar.

Now, $v_{f5} = v_{f9} = v_{f11}$ = saturated volume of water

Figure 4.17 *Regenerative feed water heating with closed heaters*

Figure 4.18 *T-s diagram for regenerative Rankine cycle with closed heaters*

Net work done, $w_{\text{net}} = w_t - \sum w_{\text{p}}$

Heat added in the boiler, $q_a = h_1 - h_{f13}$

Neglecting heat losses and considering heat balance at feed heaters, we have

$m_2\ (h_3 - h_{f9}) = (1 - m_1 - m_2)\ (\text{h}_{f9} - h_{f\,5})$

and $m_1\ (h_2 - h_{f11}) = (1 - m_1 - m_2)\ (h_{f11} - h_{f9})$

Neglecting enthalpy increase due to

pump,

$h_{f6} = h_{f5}$

also $h_{f7} = h_{f9}$

and $h_{f8} = h_{f11}$

The condition of feed water h_{13} entering the boiler is given by,

$$(1 - m_1 - m_2)\, h_{f11} + m_2 h_{f9} + m_1 h_{f11} = 1 \times h_{13}$$

or $h_{13} = (1 - m_2)\, h_{f11} + m_2 h_{f9}$

Example 4.7

A boiler feeds a turbine at 56 bar and 600°C. Before being passed on

to the condenser at 30°C, the steam is bled for regenerative feed heating at 6.5 bar. For an ideal regenerative cycle and 1kg/s of throttle steam, determine (a) the amount of bled steam, (b) net work done, and (c) the ideal efficiency of the cycle.

Solution

The *T-s* diagram is shown in Fig. 4.19.

At p_1 = 56 bar, 600°C from steam tables,

h_1 = 3659.5 kJ/kg, s_1 = 7.2011 kJ/kg.K

At 6.5 bar, h_2 = 2987 kJ/kg

and $t_2 = 265°C$

$s_3 = s_1 = s_{f3} + x_3\ s_{gh3}$

At 30°C, $s_{f3} = 0.4369$ kJ/kg.K, $s_{fg3} = 8.01674$ kJ/kg.K

or $7.2011 = 0.436 + x_3 \times 8.0164$

or $x_3 = 0.8438$

$h_3 = h_{f3} + x_3 h_{gh3} = 125.79 + 0.8438 \times 2430.5 = 2176.6$ kJ/kg

At 6.5 bar, $h_{f6} = 684.26$ kJ/kg, $v_{f6} = 0.001104\ m^3/kg$

$h_7 = h_{f6} + v_{f6}\ (p_1 - p_2) \times 10^2 = 684.26 + 0.001104\ (56 - 6.5) \times 10^2$

$= 684.26 + 5.47 = 689.72$ kJ/kg

$w_{p2} = 5.47$ kJ/kg

$v_{f4} = 0.001004$ m^3/kg

p_4 corresponding to 30°C = 0.042461 bar

$w_{p1} = v_{f\,4}\,(p_2 - p_4) = 0.001004(6.5 - 0.042461) \times 10^2$

$= 0.648$ kJ/kg

$\sum w_p = w_{p1} + w_{p2} = 0.648 + 5.47 = 6.118$ kJ/kg

Figure 4.19 *Regenerative Rankine cycle with a single heater*

1. Neglecting pump work, for heat balance of heater,

$$m_1\,(h_2 - h_{f6}) = (1 - m_1)\,(h_{f7} - h_{f4})$$

2. $w_{\text{net}} = w_t - \sum w_p = (h_1 - h_2) + (1 - m_1)\,(h_2 - h_3) - \sum w_p$

$= (3659.5 - 2987) + (1 - 0.195)\,(2987 - 2176.6) - 6.118$

$= 1318.754$ kJ/kg

3. Heat added, $q_a = (h_1 - h_7) = 3659.5 - 689.72 = 2969.78$ kJ/kg

Cycle efficiency,

Example 4.8

Determine the improvement of efficiency which would results if a single stage of regenerative feed heating were added to a steam cycle having terminal conditions of 16 bar, 300°C and 0.05 bar. The steam for feed heating is to be extracted at 1.16 bar. The drain from the heat exchanger at saturation temperature of bled steam pressure is returned to the system at a point downstream from the heat exchanger, the feed water getting heated to saturation

temperature at pressure of bled steam. Neglect change in enthalpy due to liquid pump work.

Solution

The regenerative cycle and *h-s* diagram is shown in Fig. 4.20(a) and (b).

From the Mollier diagram,

$$h_1 = 3035 \text{ kJ/kg}, h_2 = 2575 \text{ kJ/kg}, h_3 = 2098 \text{ kJ/kg}$$

From steam tables, $h_{f1} = 856.6$ kJ/kg, $h_{f2} = 475.4$ kJ/kg, $h_{f3} = 137.8$ kJ/kg

For heat balance of heater,

$$m\,(h_2 - h_{f3}) = (h_{f2} - h_{f3})$$

Figure 4.20 *Single stage regenerative heating Rankine cycle: (a) Schematic diagram, (b) Mollier diagram*

or

$$w_{\text{net}} = (h_1 - h_2) + (1 - m)(h_2 - h_3)$$

$$= (3035 - 2575) + (1 - 0.138)(2575 - 2098) = 871 \text{ kJ/kg}$$

$$q_a = h_1 - h_{f2} = 3035 - 475.4 = 2559.6 \text{ kJ/kg}$$

Thermal efficiency with regeneration,

Without feed water heating,

$$w_{\text{net}} = h_1 - h_3 = 3035 - 2098 = 937 \text{ kJ/kg}$$

$$q_a = h_1 - h_{f3} = 3035 - 137.8 =$$

2897.2 kJ/kg

Thermal efficiency without regeneration,

Improvement in thermal efficiency =

Example 4.9

A regenerative cycle with three-stage bleed heating works between 30 bar, 450°C, and 0.04 bar. The bleed temperatures are chosen at equal temperature ranges. Determine the efficiency of the cycle. Neglect the pump work.

Solution

The schematic diagram of the cycle, T-s, and h-s diagrams are shown in Fig. 4.21(a) to (c).

At $p = 0.04$ bar, $t_s = 30°C$

Temperature range = 450 – 30 = 420°C

Equal temperature difference =

The temperatures of bled steam are: 450 – 105 = 345°C, 240°C, 135°C

From Mollier chart,

$h_1 = 3340$ kJ/kg, $h_2 = 3150$ kJ/kg, $h_3 = 2935$ kJ/kg,

$h_4 = 2720$ kJ/kg, $h_s = 2120$ kJ/kg

The pressure at 2, 3, 4 points are noted from the *h*-*s* chart. Then from steam tables,

h_{f1} = 1008.35 kJ/kg, h_{f2} = 837.45 kJ/kg, h_{f3} = 684.0 kJ/kg,

h_{f4} = 517.62 kJ/kg, h_{f5} = 121.4 kJ/kg

For heat balance of heaters,

$$m_3 (h_4 - h_{f4}) = (1 - m_1 - m_2 - m_3)(h_{f4} - h_{f5})$$

$$m_2 (h_3 - h_{f3}) = (1 - m_1 - m_2 - m_3)(h_{f3} - h_{f4})$$

$$m_1 (h_2 - h_{f2}) = (1 - m_1)(h_{f2} - h_{f3})$$

$\therefore m_3 (2720 - 517.62) = (1 - m_1 -$

$m_2 - m_3)$ (517.62 – 121.4)

or 2202.38 $m_3 = (1 - m_1 - m_2 - m_3) \times 396.22$

m_2 (2935 – 684) = $(1 - m_1 - m_2)$ (684 – 517.62)

Figure 4.21 *Regenerative Rankine cycle with three-stage bleeding: (a) Schematic diagram, (b) T-s diagram, (c) h-s diagram*

or 2251 $m_2 = (1 - m_1 - m_2) \times 166.38$

m_1 (3120 – 837.45) = $(1 - m_1)$ (837.45 – 684)

or 2282.55 $m_1 = (1 - m_1) \times 153.45$

Solving the above three equations, we get

$m_1 = 0.063$ kg, $m_2 = 0.0645$ kg, $m_3 = 0.1332$ kg

Net work done, $w_{net} = 1 \times (h_1 - h_2) + (1 - m_1)(h_2 - h_3) + (1 - m_1 - m_2)(h_3 - h_4) + (1 - m_1 - m_2 - m_3)(h_4 - h_5)$

$= (3340 - 3120) + (1 - 0.063)(3120 - 2935) + (1 - 0.063 - 0.0645)(2935 - 2720) + (1 - 0.063 - 0.0645 - 0.13327)(2720 - 2120) = 1024.6$ kJ/kg

Heat added, $q_a = h_1 - h_{f2} = 3340 - 837.45 = 2502.55$ kJ/kg

Thermal efficiency of the cycle,

Reheating is mainly adopted to gain advantage of drier steam in the low pressure stages, and regeneration is adopted for increasing the thermal efficiency of the cycle. Therefore, it is advisable to execute cycle adopting both reheat and regeneration. Figure 4.22(a) shows the schematic diagram of such a cycle with two-stage regeneration and one-stage reheating at first bleeding point. The same cycle is represented on *T-s* diagram in Fig. 4.22(b).

Figure 4.22 *Reheat-regenerative cycle: (a) Schematic diagram, (b) T-s diagram*

At point 2, m_1 kg of steam is extracted from the turbine and is passed on to open heater 2 and the remaining $(1 - m_1)$ kg steam goes to reheater. The steam from the reheater comes back to

the turbine at state point 3. Thereafter, $(1 - m_1)$ kg of steam expands from state point 3 to state point 4. At point 4, again m_2 kg of steam is extracted and passed on to open heater 1. From state point 4, $(1 - m_1 - m_2)$ kg of steam expands to condenser pressure state point 5, and further condenses in the condenser to water down to state point 6. Neglecting pump work, net work output from the prime mover is given by,

$$w_{out} = (h_1 - h_2) + (1 - m_1)(h_3 - h_4) + (1 - m_1 - m_2)(h_4 - h_5)$$

Net work of the cycle, $w_{net} = w_{out} - \sum w_p$

where $\sum w_p = w_{p1} + w_{p2} + w_{p3}$

$= (1 - m_1 - m_2)(h_{b6} - h_6) + (1 - m_1)$
$(h_{b7} - h_7) + (h_{b8} - h_8)$

$\simeq v_{f6}(p_1 - p_6) \times 10^2$ kJ/kg

Heat added, $q_a = (h_1 - h_{f8}) + (1 - m_1)$
$(h_3 - h_2) - \sum w_p$

Thermal efficiency,

4.7 ❒ PROPERTIES OF AN IDEAL WORKING FLUID

An ideal working fluid should have the following favourable properties:

1. Plentiful and cheap availability
2. Non-toxic
3. Chemical stability
4. Non-corrosiveness
5. High critical temperature
6. Saturation pressure should be above atmospheric pressure to guard against leakage

4.8 ❒ BINARY VAPOUR CYCLES

In a binary vapour cycle, two working

substances are used—one with good characteristics at high temperature and the other with good at low temperature characteristics. Figures 4.23(a) and (b) show two Rankine cycles in a combined arrangement. The heat is rejected from the high temperature cycle (mercury cycle) as energy supplied to the low temperature cycle (steam cycle). This occurs in a heat exchanger which functions as the condenser for mercury cycle and boiler for the water cycle. Specific enthalpy increase of water as it passes through the heat exchanger is several times the magnitude of the specific enthalpy decrease of mercury. Thus, the mass of mercury circulated in mercury cycle is several times more than mass of water in the water cycle.

Binary vapour power cycles have a great advantage of its operation at high average heat addition temperature than conventional water cycles and give higher thermal efficiency.

Figure 4.23 *Binary vapour cycles: (a) Schematic diagram, (b) Mercury and steam cycles*

Mercury cycle A–B–C–D–A is a simple Rankine cycle using saturated vapour. Heat is supplied to mercury in the process D–A. Mercury expands in process A–B and is condensed in process B–C. Mercury-saturated liquid is compressed in feed pump from condenser pressure to boiler pressure in process C–D. Thus, the Rankine cycle on mercury is completed.

Heat rejected during condensation in the

mercury condenser is supplied to water to transform water into saturated steam in process 4–5. Process 4–5 may take place partly in the economiser located in the exhaust gases of mercury boiler and partly in the mercury condenser. The saturated vapour is superheated in the superheater located in the mercury boiler furnace process 5–1. Ideal process 4–4′occurs in the economiser and 4′–5 in the mercury condenser. Superheated steam expands in the turbine, process 1–2, and then condenses in the steam condenser, process 2–3. The condensate or feed water is then pumped by process 3–4, and heated till it is saturated liquid in the economiser as shown by process 4–4′, before going to the mercury condenser -

steam boiler where enthalpy of evaporation is absorbed.

Let m = flow rate of mercury in the mercury cycle for 1 kg steam being circulated in steam cycle.

Heat added, $Q_a = m\ (h_A - h_D) + (h_1 - h_5) + (- h_4)$

Heat rejected, $Q_r = (h_2 - h_3)$

Workdone by steam and mercury turbines,

$W_t = m\ (h_A - h_B) + (h_1 - h_2)$

Workdone on the feed pumps for mercury liquid and water,

$$W_p = m\,(h_D - h_C) + (h_4 - h_3)$$

Thermal efficiency of binary cycle,

The mercury cycle is known as topping cycle and the steam cycle is known as bottoming cycle.

Example 4.10

A binary vapour cycle consists of two ideal Rankine cycles with mercury and steam as working substances. The following data is given:

Saturated mercury vapour pressure at entry to turbine = 4.5 bar

Mercury vapour exit pressure from turbine = 0.04 bar

Saturated steam generated in mercury condenser pressure = 15 bar

Steam superheated temperature in superheater in mercury boiler = 300°C

Condensate water pressure pumped through economiser located in exhaust flue of mercury boiler to saturation temperature = 15 bar

Isentropic efficiency of mercury turbine = 0.85

Isentropic efficiency of steam

turbine = 0.88

Calculate (a) the overall thermal efficiency of the cycle and (b) the rate of flow of mercury turbine in kg/h.

Properties of saturated mercury

Figure 4.24 *Hg – H_2O binary vapour cycle*

Solution

Refer to Fig. 4.24.

Mercury cycle

h_A = 3559 kJ/kg, p_A = 4.5 bar, t_A = 450°C, s_A = 0.5397 kJ/kg.K

Now $s_A = s_{fB} + x_B (s_{gB} - s_{fB})$

or $0.5397 = 0.0808 + x_B (0.6925 - 0.0808)$

or $x_B = 0.75$

$h_B = h_{fB} + x_B (h_{gB} - h_{fB}) = 29.98 + 0.75 (329.85 - 29.98) = 254.88$ kJ/kg

$h_B' = h_A - (h_A - h_B) \eta_{t1} = 355.98 - (355.98 - 254.88) \times 0.85 = 270.04$ kJ/kg

$p_B' = 0.04$ bar, $t_B' = 216.9°C$

$h_c = h_{fc} = 29.98$ kJ/kg, $t_c = 216.9°C$

$h_D = h_c + v_{fc} (p_D - p_C) \times 10^2 = 29.98 + 76.5 \times 10^{-6} (4.5 - 0.04) \times 10^2 = 30.0142$ kJ/kg

$w_{p1} = v_{fc} (p_D - p_C) \times 10^2 = 0.03412$ kJ/kg

Steam cycle

$p_1 = 15$ bar, $t_1 = 300°C$, $h_1 = 3038.9$ kJ/kg, $s_1 = 6.9207$ kJ/kg. K

$p_2 = 0.05$ bar, $h_{f2} = 137.77$ kJ/kg, $h_{g2} = 2561.6$ kJ/kg, $s_{f2} = 0.4763$ kJ/kg.K,

$h_5 = 2789.9$ kJ/kg, $h_4' = 844.66$ kJ/kg

$h_{fg2} = 2423.8$ kJ/kg, $s_{g2} = 8.3960$ kJ/kg.K, $v_{f2} = 0.0010052$ m^3/kg

$s_1 = s_2 = s_{f2} + x_2\, s_{fg2}$

or $6.9207 = 0.4763 + x_2 (8.3980 - 0.4763)$

or $x_2 = 0.8137$

$h_2 = h_{f2} + x_2 h_{fg2} = 137.77 + 0.8137 \times 2423.8 = 2110$ kJ/kg

$h_2' = h_1 - (h_1 - h_2)\, \eta_{t2} = 3038.9 - (3038.9 - 2110) \times 0.88 = 2221.47$ kJ/kg

$h_3 = h_{f3} = 137.77$ kJ/kg

$w_{p2} = v_{f3} (p_4 - p_3) \times 10^2 = 0.0010052 (15 - 0.05) \times 10^2 = 1.503$ kJ/kg

$$h_4 = h_3 + w_{p2} = 137.77 + 1.503 = 139.27 \text{ kJ/kg}$$

Total heat added, $(q_a)_{\text{total}} = m_{\text{Hg}} (h_{\text{A}}$

$- h_D) + 1\ [(h_1 - h_5) + h_{4'} - h_4)]$

$$= 8.103\ (355.98 - 30.0142) + 1\ [(3038.9 - 2789.9) + (844.66 - 139.27)] = 3595.7 \text{ kJ/kg}$$

Work done, $(w_t)_{Hg} = m_{Hg}\ (h_A - h_{B'})$ $= 8.103\ (355.98 - 270.04) = 696.37$ kJ/kg of steam

$(w_t)_{steam} = 1 \times (h_1 - \ ) = 1 \times (3038.9 - 2221.47) = 817.43$ kJ/kg

$(w_t)_{total} = 696.37 + 817.43 = 1513.8$ kJ/kg

Neglecting pump work being very small,

Considering pump work,

In the Rankine cycle, the heat rejected is considered to be of no value. However, this heat can be supplied by steam, for the heating of buildings, and heating required by many industrial processes. Thus, the functions of power generation and heating can often be combined effectively. This combination is often called *cogeneration*.

In such a cycle, the process heater replaces the condenser of an ordinary Rankine cycle. The pressure of the exhaust from the turbine is the saturation pressure corresponding to the temperature desired in the process heater. Such a turbine is called *back pressure turbine*. The cycle for a single

plant carrying varying heat and power loads is shown in Fig. 4.25.

Figure 4.25 *Combined power and heating cycle*

When the power load is zero, all the steam passes through the pressure reducing valve (PRV) and none through the turbine. When heating load is zero, the steam expands through the turbine and flows into the condenser. In order to meet heating and power loads, the steam flow distribution is determined as follows:

The steam is extracted from the turbine at point 2 where the pressure is held constant by an automatic valve arrangement. The flow rate in the extraction line is computed from energy balance on the heating system as:

The power produced by this steam flowing through the turbine from the extraction point is,

$$W = m_2 (h_1 - h_2)$$

Now $h_1 = h_6$ and $h_2 = h_1 - w$, so that $h_6 > h_2$.

The steam distribution to compute m_2, m_3 and m_6 must satisfy the First Law of Thermodynamics for power required.

Example 4.11

In a binary-vapour cycle, the steam cycle operates between 30 bar, 0.07 bar and uses a superheated temperature of 350°C. The mercury

cycle works between 12.68 bar and 0.07 bar. The mercury vapour entering the turbine is in a dry saturated condition. Calculate the efficiency of the combined cycle assuming expansion in both cycles to be isentropic. Data for mercury is given below:

Solution

The binary vapour cycle is shown in Fig. 4.26.

Mercury cycle: $s_a = s_b = s_{fb} + x_b (s_{gb} - s_{fb})$

$0.50185 = 0.08548 + x_b (0.662906 - 0.08548)$

or $x_b = 0.721$

Figure 4.26 *Binary vapour cycle neglecting pump work*

$h_b = h_{fb} + x_b\,(h_{gb} - h_{fb})$

$= 32.395 + 0.721\,(326.667 - 32.395) = 244.565$ kJ/kg of Hg

Isentropic work done, $w_m = h_a - h_b = 360.734 - 244.565 = 116.169$ kJ/kg Hg

Heat rejected, $q_{rm} = h_b - h_{fb} = 244.565 - 32.395 = 212.17$ kJ/kg Hg

Heat supplied, $q_{sm} = h_a - h_{fb} = 360.734 - 32.395 = 328.339$ kJ/kg Hg

Steam cycle: $h_1 = 3115.3$ kJ/kg, $s_1 = 6.743$ kJ/kg.K, $h'_1 = 2804.2$ kJ/kg

$$s_{f2} = 0.5589 \text{ kJ/kg.K}, s_{fg2} = 7.7179 \text{ kJ/kg.K}, h_{f3} = 163.40 \text{ kJ/kg}$$

$$s_1 = s_2 = s_{f2} + x_2\, s_{fg2}$$

or $6.743 = 0.5589 + x_2 \times 7.7179$

or $x_2 = 0.80127$

$$h_2 = h_{f2} + x_2\, h_{fg2} = 163.40 + 0.80127 \times 2409.1 = 2093.733 \text{ kJ/kg}$$

Work done, $w_s = h_1 - h_2 = 3115.3 - 2093.733 = 1021.567$ kJ/kg

Heat supplied, $q_{ss} = h_1 = h_1' = 3115.3 - 2804.2 = 311.1$ kJ/kg

Heat rejected by Hg/kg of steam = Heat received by water/kg steam

$m_m\,(h_b - h_{fb}) = 1\,(h_1' - h_{f3})$

or

Total work done per kg of steam,

$w_1 = w_s + m_m\, w_m = 1021.567 + 12.447 \times 116.161 = 2467.379$ kJ/kg steam

Heat supplied in the cycle,

$q_s = m_m\, q_{sm} + q_{ss} = 12.447 \times 328.339 + 311.1 = 4397.936$ kJ/kg steam

Cycle efficiency =

Example 4.12

In a single-heater regenerative

cycle, the steam enters the turbine at 32 bar, 350°C and the exhaust pressure is 0.12 bar. The feed water heater is a direct contact type which operates at 4.5 bar. Find (a) the efficiency and the steam rate of the cycle and (b) the increase in efficiency and steam rate as compared to Rankine cycle. Neglect pump work.

Solution

From steam tables, $h_1 = 3113.2$ kJ/kg, $s_1 = 6.712$ kJ/kg.K $= s_2 = s_3$

s_g at 4.5 bar = 6.875 kJ/kg.K

The schematic diagram is shown in Fig. 4.27(a). The processes on the

T-s and h-s diagrams are shown in Figs. 4.27(b) and (c) respectively. Since $s_2 < s_g$, state point 2 lies in the wet region.

$s_{f2} = 1.821$ kJ/kg.K, $s_{fg2} = 5.036$ kJ/kg.K

$s_1 = s_2 = s_{f2} + x_2\, s_{fg2}$

or $6.712 = 1.821 + x_2 \times 5.036$

or $x_2 = 0.9712$

$h_2 = h_{f2} + x_2\, h_{fg2} = 623.3 + 0.9712 \times 2120.6 = 2682.83$ kJ/kg

Figure 4.27 *Single stage regenerative cycle: (a) Flow schematic diagram, (b) T-s diagram, (c) h-s diagram*

$s_{f3} = 0.6963$ kJ/kg.K $s_{fg3} = 7.3900$ kJ/kg.K

$s_1 = s_3 = s_{f3} + x_3\, s_{fg3}$

or $6.712 = 0.6963 + x_2 \times 7.3900$

or $x_3 = 0.814$

$h_3 = h_{f3} + x_3\, h_{fg3} = 206.92 \div 0.814 \times 2384.1 = 2147.58$ kJ/kg

Since $w_p = 0$

$\therefore h_4 = h_{f3} = 206.92$ kJ/kg $= h_5$

$h_6 = h_{f2} = 623.3$ kJ/kg $= h_7$

Energy balance for heater gives,

$$m_1\,(h_2 - h_6) = (1 - m_1)\,(h_6 - h_5)$$

or $m_1\,(2682.83 - 623.3) = (1 - m_1)\,(623.3 - 206.92)$

or $2059.53\, m_1 = (1 - m_1) \times 416.38$

or $m_1 = 0.168$ kg

$w_t = (h_1 - h_2) + (1 - m_1)\,(h_2 - h_3)$

$= (3113.2 - 2682.83) + (1 - 0.168)$
$(2628.83 - 2147.58) = 875.698$ kJ/
kg

$q_s = h_1 - h_6 = 3113.2 - 623.3 =$
2489.9 kJ/kg

Steam rate =

Without regeneration, $w_t = h_1 - h_3 =$
$3113.2 - 2147.58 = 965.62$ kJ/kg

Increase in cycle efficiency due to

regeneration = 35.17 – 33.22 = 1.95%

Steam rate =

Increase in steam rate due to regeneration = 4.11 – 3.728 = 0.382 kg/kWh

Example 4.13

A steam power plant working on Rankine cycle is supplied with dry saturated steam at a pressure of 12 bar and exhausts into the condenser at 0.1 bar. Neglecting the pump work, calculate the cycle efficiency.

Solution

Corresponding to p = 12 bar, from steam tables we find that (see Fig. 4.3)

h_{f1} = 798.4 kJ/kg; h_{g1} = 1984.3 kJ/kg

$s_1 = s_{g1}$ = 6.519 kJ/kg°K

Corresponding to 0.1 bar, we find that

h_{f2} = 191.8 kJ/kg; h_{fg2} = 2393 kJ/kg

s_{f2} = 0.649 kJ/kg K; s_{fg2} = 7.502 kJ/kg·K

Process 1–2 is an isentropic expansion

$s_1 = s_2$

$s_1 = s_{g1} = 6.519$ kJ/kg K

$s_2 = s_{f\,2} + x_2\, s_{fg2}$

$s_2 = 0.649 + x2 \times 7.502$

$6.519 = 0.649 + 7.502\, x_2$

$x_3 = 0.783$

$h_2 = h_{f2} + h_{fg2}$

$=191.81 + 0.783 \times 2392.8 = 2065.5$ kJ/kg

$h_1 = h_{g2} = 2782.7$ kJ/kg

Example 4.14

Dry saturated steam at 10 bar is supplied to a prime mover working on Rankine cycle and the exhaust takes place at 0.2 bar. Determine the cycle efficiency, efficiency ratio, and specific steam consumption of the prime mover, if the indicated thermal efficiency is 20%. Determine the percentage change in the cycle efficiency if the steam is initially 90% dry.

Solution

The *h*-*s* diagram is shown in Fig. 4.28.

From Mollier diagram (Fig. 4.28), we have

$h_1 = 2775$ kJ/kg, $h_2 = 2150$ kJ/kg

From steam tables,

$h_{f2} = 251.5$ kJ/kg corresponding to 0.2 bar

Thermal efficiency

Efficiency ratio =

Specific steam consumption =

Percentage change in cycle efficiency if steam is initially 90% dry

From Mollier diagram, $h_1 = 2580$ kJ/kg, $h_2 = 2030$ kJ/kg (see Fig. 4.29)

Thermal efficiency

∴ Percentage change in Rankine efficiency =

Figure 4.28 *Mollier diagram for dry saturated steam*

Figure 4.29 *Mollier diagram for wet steam*

Example 4.15

The steam consumption of steam engine is 20 tonnes per shift of 8 hours when developing 220 kW. Dry and saturated steam enters the engine at 10 bar pressure and leaves at 0.1 bar pressure. Estimate the cycle efficiency and thermal efficiency of engine.

Solution

Given that m_s = 20/8 tonne/h = 2.5 tonne/h = 2500 kg/h, P = 220 kW

The *h-s* diagrams is shown in Fig. 4.30.

From steam tables corresponding to 10 bar

$h_1 = h_{g1}$ = 2778.1 kJ/kg; $s_1 = s_{g1}$ = 6.5864 kJ/kg.K

and corresponding to 0.1 bar, we find that

h_{f2} = 191.81 kJ/kg; h_{fg2} = 2392.8 kJ/kg;

s_{f2} = 0.6492 kJ/kg.K; s_{g2} = 7.5010 kJ/kg.K

Since 1–2 is an isentropic process

$s_1 = s_3$

or $6.5864 = 0.6492 + x_2 \times 7.5010$

or $x_2 = 0.791$

$\therefore h_3 = h_{f2} + x_2\, h_{fg2} = 191.81 + 0.791 \times 2392.8 = 2084.5$ kJ/kg

Cycle Efficiency (η_R) =

Thermal efficiency of engine;

Figure 4.30 *Mollier diagram for dry saturated steam*

Example 4.16

Steam at 50 bar, 400°C expands in a

Rankine engine to 0.34 bar. For 150 kg/s of steam; determine (a) the power developed, (b) the thermal efficiency neglecting the pump work, and (c) the specific steam consumption.

Solution

The *h-s* diagrams is shown in Fig. 4.31.

Figure 4.31 *Mollier diagram for superheated steam*

1. **Power developed** = $m(h_1 - h_2)$

   From Mollier diagram (Fig. 4.31), we have

   $h_1 = 3198.3$ kJ/kg, $h_2 = 2257.75$ kJ/kg

   $h_{f2} = 301.5$ kJ/kg (from steam tables)

   Power developed = 150 [3198.3 − 2257.75] = 141082 kW

2. **Thermal efficiency**
3. **Specific steam consumption** =

Example 4.17

A steam engine is supplied with dry saturated steam at 15 bar. The pressure at release is 3 bar and the back pressure is 1 bar. Determine the efficiency of modified Rankine cycle.

Solution

Consider the Mollier diagram, as shown in Fig. 4.32.

From the Mollier diagram, we have

$h_1 = 2790$ kJ/kg, $h_2 = 2510$ kJ/kg, $x_2 = 0.9$

From steam tables, corresponding to

3 bar, we have

v_{f2} =0.001073 m^3/kg, v_{g2} = 0.6058 m^3/kg

∴ Volume of steam at point 3,

$v_2 = v_{f2} + x_2 (v_{g2} - v_{f2}) = 0.001073 + 0.9 \times (0.60585 - 0.001073) = 0.545$ m^3/kg

Figure 4.32 *Mollier diagram for dry saturated steam*

From steam tables corresponding to pressure of 1 bar, we find that sensible heat of water,

h_{f4} = 417.44 kJ/kg

η_{MR} = Thermal efficiency of modified Rankine cycle

Example 4.18

Steam at a pressure of 15 bar and 250°C is first expanded through a turbine to a pressure of 4 bar. It is then reheated at constant pressure to the initial temperature of 250°C and is finally expanded to 0.1 bar. Estimate the work done per kg of steam flowing through the turbine and the amount of heat supplied during the process of reheat.

Find the work output when there is direct expansion from 15 bar to 0.1 bar without any reheat. Assume all expansion processes to be isentropic.

Solution

Given that $p_1 = 15$ bar; $T_1 = 250°C$; $p_2 = 4$ bar; $T_2 = 250°C$; $p_3 = 0.1$ bar

The reheating of steam is represented on the Mollier chart as shown in Fig. 4.33. From the chart, we find that,

$h_1 = 2930$ kJ/kg; $h_2 = 2660$ kJ/kg; $h_3 = 2965$ kJ/kg; $h_4 = 2345$ kJ/kg; and $h_5 = 2130$ kJ/kg

From steam tables, corresponding to a pressure of 0.1 bar, we find that sensible heat of water at D,

$h_{f4} = h_{f5} = 191.81$ kJ/kg

Workdone per kg of steam:

We know that workdone per kg of steam,

$w = (h_1 - h_2) + (h_3 - h_4)$

$= (2930 - 2660) + (2965 - 2345) = 890$ kJ/kg

Heat supplied during the process of reheat:

We know that the heat supplied during the process of reheat,

h = Heat supplied between 2 and 3

$= (h_3 - h_2) - (h_{f4}) = (2965 - 2660) - 191.81 = 113.2$ kJ/kg

Figure 4.33 *Mollier diagram for superheated steam with reheating*

Work output when the expansion is direct:

The direct expansion from 15 bar to 0.1 bar is shown by the line AE in Fig. 4.33. We know that work output

= Total heat drop = $h_1 - h_5$ = 2930 – 2130 = 800 kJ/kg

Example 4.19

In a thermal plant, steam is supplied at a pressure of 30 bar and temperature of 300°C to the high pressure side of steam turbine where it is expanded to 5 bar. The steam is

then removed and reheated to 300°C at a constant pressure. It is then expanded to the low pressure side of the turbine to 0.5 bar. Find the efficiency of the cycle with and without reheating.

Solution

Given that $p_1 = 30$ bar; $T_1 = 300$°C; $p_2 = 5$ bar; $T_2 = 300$°C; $p_3 = 0.5$ bar

Efficiency of the cycle with reheating:

The reheating of steam is represented on the Mollier chart as shown in Fig. 4.34.

From the chart, we find that

$h_1 = 2990$ kJ/kg; $h_2 = 2625$ kJ/kg; $h_3 = 3075$ kJ/kg; $h_4 = 2595$ kJ/kg; and $h_5 = 2280$ kJ/kg

From steam tables, corresponding to a pressure of 0.5 bar, we find that sensible heat of water at D,

$$h_{f4} = h_{f5} = 340.47 \text{ kJ/kg}$$

Figure 4.34 *Mollier diagram for superheated steam with reheating*

We know that efficiency of the cycle with reheating,

Efficiency of the cycle without reheating:

We know that efficiency of the cycle without reheating

Example 4.20

Steam supplied to a 10 MW turbo-alternator at 40 bar and 400°C. The auxiliaries consume 7% of the output. The condenser pressure is 0.05 bar and condensate is sub-cooled to 30°C. Assuming the boiler efficiency as 85%, the relative efficiency of turbine as 80%, and the mechanical efficiency of the alternator as 95%, determine (a) the steam consumption per hour, (b) the overall efficiency of the plant, and (c) the quality of steam at exit from turbine.

Solution

The *h-s* diagrams is shown in Fig. 4.35.

1. **From Mollier chart,**

   $h_1 = 3215$ kJ/kg, $h_2 = 2064$ kJ/kg

   Therefore, $(h_1 - h_2) = 3215 - 2064 = 1151$ kJ/kg

   Thus actual enthalpy drop,

   $(h_1 - h_2) = \eta_{\text{relative}} \times (h_1 - h_2)$

   $= 0.8 \times 1151 = 920.8$ kJ/kg

   Input to the alternator =

   Figure 4.35 *Mollier diagram for superheated steam*

   Therefore, steam consumption per hour,

2. **Heat supplied to the boiler/hour** =

   where h_{f2} = enthalpy of liquid at saturation at 0.05 bar from steam tables

   The auxiliaries consume 7% of the output.

   Therefore, useful output = $(10{,}000 \times 0.93) = 9300$ kW $= 9300 \times 60 \times 60$ kJ/h

Thus overall efficiency =

3. **The dryness fraction at exit**, that is, B′ is read at $h_{2'} = 3215 - 920.8 = 2294.2$ kJ on 0.05 bar line.

Thus, $x_2' = 0.889$

Example 4.21

Two turbines, A and B, operate with steam at an initial pressure of 100 bar and temperature of 500°C. In each turbine, the steam is expanded in a high pressure turbine to 8.5 bar with efficiency ratio 80%. In turbine A, the expansion is further continued in low pressure turbine from 8.5 bar to 0.035 bar with efficiency ratio 0.75. In turbine B, steam is reheated after expansion in the high pressure turbine and is then fed to the low pressure unit at 7 bar

and 500°C, after which it expands to 0.035 bar with efficiency ratio 0.85. Compare the two power cycle with respect to (a) thermal efficiency and (b) steam consumption for a full load output of 50,000 kW.

Solution

Turbine A: The expansion is shown on *h-s* chart in Fig. 4.36(a)

$h_a = 3377$ kJ/kg and $h_b = 2753$ kJ/kg

$h_c = h_d = 3377 - 0.8\ (3377 - 2753) = 2878$ kJ/kg

$h_e = 2060$ kJ/kg

$h_f = h_g = 2878 - (2878 - 2060) \times$

0.75 = 2264.5 kJ/kg

Total work done = $(h_a - h_d) + (h_d - h_g) = (h_a - h_g)$

$$= 3377 - 2264.5 = 1112.5 \text{ kJ/kg}$$

Heat supplied = $h_a - h_g$

where h_g is the sensible heat of steam at 0.035 bar = 110 kJ

∴ Heat supplied = 3377 − 110 = 3267 kJ

Thermal efficiency =

Total work to be done = 53,000 kW = 50,000 kJ/s

$$= 50{,}000 \times 3600 \text{ kJ/h}$$

Figure 4.36 *Mollier diagrams: (a) h-s diagram for Turbine A, (b) h-s diagram for Turbine B*

∴ Steam consumption =

Turbine B: The expansion is shown on h-s chart in Fig. 4.36(b)

h_a = 3377 kJ/kg: h_b = 2753 kJ/kg; h_c = h_d = 2878 kJ/kg

h_e = 3490 kJ/kg; h_f = 2385 kJ/kg.

h_h = 3490 – (3490 – 2315) 0.85 = 2551 kJ/kg

Total work done = (3377 – 2878) + (3490 – 2551) = 1438 kJ/kg

Total heat supplied = (3377 – 110) + (3490 – 2878) = 3879 kJ/kg.

∴ Thermal efficiency

Steam consumption =

Example 4.22

In a regenerative cycle, having one feed water heater, the dry saturated steam is supplied from the boiler at a pressure of 30 bar and the condenser pressure is 1 bar. The steam is bled at a pressure of 5 bar. Determine the amount of bled steam per kg of steam supplied and the efficiency of the cycle. What would be the efficiency without regenerative feed heating? Determine the percentage increase in efficiency due to regeneration.

Solution

Given that $p_1 = 30$ bar; $p_3 = 1$ bar; $p_2 = 5$ bar

From Mollier diagram, as shown in Fig. 4.37, we find that

Enthalpy of steam at 30 bar, $h_1 =$ 2800 kJ/kg

Enthalpy of steam at 5 bar, $h_2 =$ 2460 kJ/kg

Enthalpy of steam at 1 bar, $h_3 =$ 2220 kJ/kg

From steam tables, we also find the enthalpy or sensible heat of water at 5 bar.

Figure 4.37 *Mollier diagram for dry saturated steam*

$h_{f2} = 640.21$ kJ/kg

and enthalpy or sensible heat of water at 1 bar,

$h_{f3} = 417.44$ kJ/kg

Amount of bled steam per kg steam supplied

We know that amount of bled steam per kg of steam supplied,

Efficiency of the cycle: We know that efficiency of the cycle,

Efficiency of the cycle without regenerative feed heating: We know that efficiency of the cycle,

Percentage increase in efficiency due to regeneration: We know that percentage increase in efficiency due to regeneration

Example 4.23

In a steam turbine plant, the steam is generated and supplied to the turbine at 50 bar and 370°C. The condenser pressure is 0.1 bar and the steam enters the condenser with dryness fraction of 0.9. Two feed water heaters are used, the steam in the heaters being bled at 5 bar and 0.5 bar. In each heater, the feed water is heated to saturation temperature of the bled steam. The

condensate is also pumped at this temperature into the feed line immediately after the heater. Find the masses of the steam bled in the turbine per one kg of steam entering the turbine. Assuming the condition line for the turbine to be straight, calculate the thermal efficiency of the cycle.

Solution

Given that $p_1 = 50$ bar; $T_1 = 370°C$; $p_4 = 0.1$ bar; $x_4 = 0.9$; $p_2 = 5$ bar; $p_3 = 0.5$ bar

First, let us draw the Mollier diagram and condition line for the cycle, as shown in Fig. 4.38. From this diagram, we find that,

$h_1 = 3110$ kJ/kg, $h_2 = 2780$ kJ/kg, $h_3 = 2510$ kJ/kg, $h_4 = 2350$ kJ/kg

From steam tables, we also find that

$h_{f2} = 640.21$ kJ/kg (at 5 bar)

$h_{f3} = 340.47$ kJ/kg (at 0.5 bar)

$h_{f4} = 191.81$ kJ/kg (at 0.1 bar)

Mass of steam bled in the turbine:

We know that mass of steam bled at 2.

and mass of steam bled at 3.

Thermal efficiency of the cycle:

We know that work done from 1 to 2 per kg of feed water

$= h_1 - h_2 = 3110 - 2780 = 330$ kJ/kg

Similarly, work done from 2 to 3 per kg of feed water

$= (1 - m_1)(h_2 - h_3) = (1 - 0.123)(2780 - 2510)$ kJ/kg

$= 236.8$ kJ/kg

and work done from 3 to 4 per kg of feed water

$= (1 - m_1 - m_2)(h_3 - h_4) = (1 - 0.123 - 0.056)(2510 - 2350)$ kJ/kg

= 131.4 kJ/kg

Figure 4.38 *Mollier diagram*

∴ Total work done = 330 + 236.8 + 131.4 = 698.2 kJ/kg

Heat supplied = $h_1 - h_{f2}$ = 3110 – 640.1 = 2469.9 kJ/kg

∴ Thermal efficiency of cycle,

Example 4.24

In the Rankine cycle, steam leaves the boiler and enters the turbine at 4 MPa and 400°C. The condenser pressure is 10 kPa. Neglecting pump work, determine the cycle efficiency and the Carnot efficiency for the

same temperature limits.

Solution

Refer to Fig. 4.39. Rankine cycle is represented by 1–2–3–4 ignoring pump work, Carnot cycle for the same temperature limits may be reproduced by 1–2′–3′–4′.

Carnot efficiency is given by

(It may be noted that Carnot efficiency is independent of the working substance.)

From steam tables, h_1 at 4MPa and 400°C = 3213.5 kJ/kg, s_1 = 6.7689 kJ/kg.K

At 10 kPa

$h_f = 191.81$ kJ/kg, $h_{fg} = 2392.8$ kJ/kg, $s_f = 0.6492$ kJ/kg.K, $s_{fg} = 7.5010$ kJ/kg.K

Also $s_1 = s_2$

or $6.7689 = 0.6492 + x_2 \times 7.5010$

or

$\therefore h_2 = h_{f3} + x_2 h_{fg} = 191.81 + 0.816 \times 2392.8 = 2144.33$ kJ/kg

Thermal efficiency $\eta_{\text{Rankine}} =$

Figure 4.39 *T-s diagram for Rankine cycle*

Example 4.25

A steam power plant has the range of operation from 40 bar dry saturated to 0.05 bar. Determine (a) the cycle efficiency and (b) the work ratio and specific fuel consumption for (i) Carnot cycle and (ii) Rankine cycle.

Solution

From steam tables:

At 40 bar, $t_g = 250.4°C$, $v_f = 0.001252\ m^3/kg$, $v_g = 0.049778\ m^3/kg$, $h_f = 1087.29\ kJ/kg$, $h_g = 2801.40\ kJ/kg$, $s_g = 6.07\ kJ/kg.K$

At 0.05 bar, $t_g = 32.88°C$, $v_f = 0.0010005\ m^3/kg$, $v_g = 28.193\ m^3/kg$, $h_f = 137.79\ kJ/kg$

h_{fg} = 2423.7 kJ/kg, s_f = 0.4763 kJ/kg.K, s_{fg} = 7.9187 kJ/kg.K

1. Refer of Fig. 4.40. for Carnot cycle analysis on *T-s* plane.

   Cycle efficiency =

   (*Note:* Processes 1–2 and 3–4 are reversible adiabatic.)

   Also $s_1 = s_2 = s_f + x_2 s_{fg}$

   or

   Also $s_4 = s_3 = s_f + x_3 s_{fg}$

   or

   Figure 4.40 *Carnot cycle on T-s plot*

   Figure 4.41 *Rankine cycle on T-s plot*

   $\therefore h_2 = 137.79 + 0.706 \times 2423.7 = 1848.9$ kJ/kg.

   and $h_3 = 137.79 + 0.293 \times 2423.7 = 847.9$ kJ/kg.

   $\therefore \eta_{\text{Carnot cycle}} =$

   Alternatively,

2. Refer to Fig. 4.41 for Rankine cycle analysis

Pump work = $h_M - h_{f3}$

$= 0.001005 \times 39.95 \times 10^2 = 4.015$ kJ/kg

Therefore, net work

$w_{net} = h_1 - h_2 - w_{pump} = 2801.4 - 1848.9 - 4.015 = 948.485$ kJ/kg

and $\eta_{\text{Rankine cycle}} =$

$= 0.3566$ or 35.66%

1. b. Also work ratio =

Specific steam consumption/kWh

Example 4.26

Steam at 50 bar, 400°C expands in a steam tubine to 0.34 bar. For 150 kg/s of steam; determine (a) the power developed, (b) the thermal efficiency, and (c) the specific steam

consumoption (i) for the Rankine cycle and (ii) for the Rankine engine. For an actual engine with same specifications, the brake steam rate is 4.75 kg/kWh and the driven electric generator has an electro mechanical efficiency of 94%.

Determine (a) the brake thermal efficiency, (b) the internal efficiency (expansion efficiency), (c) the power in kW, and (d) the exhaust dryness fraction of steam.

Solution

Refer to Fig. 4.42.

At 50 bar and 400°C

$h_1 = 3195.6$ kJ/kg, $s_1 = 6.6458$ kJ/kg.K, $v_1 = 0.05781$ m^3/kg

At 0.34 bar, $h_f = 301.48$ kJ/kg, $h_{fg} = 2328.9$ kJ/kg, $s_f = 0.9795$ kJ/kg.K, $s_{fg} = 6.7471$ kJ/kg.K

And $s_1 = s_2$ for ideal expansion

$6.6458 = 0.9795 + x_2 \times 6.7471$

or $x_2 = 0.84$

Thus $h_2 = h_{f3} + x_2 h_{fg} = 301.43 + 0.84 \times 2328.9 = 2257.756$ kJ/kg

1. For cycle w_{pump} = Pump work = $h_M - h_{f3}$
    1. Power developed = Steam per sec × ($h_1 - h_2 - w_{pump}$)

       = [150 (3195.6 – 2257.756 – 5.0857)] = 139913.7 kW

    2. $\eta_{Rankine}$ =

Figure 4.42 *Rankine cycle on T-s plot*

3. Steam rate or specific consumption =

2. For engine

    1. Power developed = Steam per sec × ($h_1 - h_2$) = 150 × (3195.6 – 2257.756)

    $$= 140676.6 \text{ kW}$$

    2. $\eta_{\text{Rankine engine}}$ =
    3. Steam rate or specific steam consumption

3. Actual engine

Brake specific steam consumption = 4.75 kg/kWh

$\eta_{\text{brake thermal}}$ =

Expansion efficiency =

Actual enthalpy drop = 0.808 × Isentropic enthalpy drop

= 0.808 (3195.6 – 2257.756) = 757.8 kJ/kg

Thus, $h'_2 = h_1 - 757.8 = 3195.6 - 757.8 = 2437.8$ kJ/kg

Also, $h'_2 = h_{f2} + xh_{fg2}$

Thus,

Example 4.27

In a steam power plant operating on Rankine cycle, the steam enters the turbine at 70 bar and 550°C with a velocity of 30 m/s. It discharges to the condenser at 0.20 bar with a velocity of 90 m/s. If the steam flow rate is 35 kg/s find the thermal efficiency and the net power produced. Neglect pump work.

[IES, 1992]

Solution

Given that $p_1 = 70$ bar, $t_1 = 550°C$, $c_1 = 30$ m/s, $p_2 = 0.20$ bar, $c_2 = 90$ m/s, $\dot{m}_s = 35$ kg/s

The simple Rankine cycle is shown in Fig. 4.43(a) and *T*-*s* diagram in Fig. 4.43(b).

Figure 4.43 *Rankine cycle: (a) Schematic diagram, (b) T-s diagram*

From steam tables, we have

At $p_1 = 70$ bar and $t_1 = 550°C$, $h_1 = 3530.9$ kJ/kg, $s_1 = 6.9486$ kJ/kg.K

At $p_2 = 0.2$ bar, $h_{f2} = 251384$ kJ/kg, $h_{fg2} = 2358.3$ kJ/kg, $s_{f2} = 0.8319$ kJ/kg.K, $s_{fg2} = 7.0766$ kJ/kg.K

Now $s_1 = s_2 = s_{f2} + x_2\, s_{fg2}$

or $6.9486 = 0.8319 + x_2 \times 7.0766$

or $x_2 = 0.8644$

$\therefore h_2 = h_{f2} + x_2\, h_{fg2}$

$= 251.38 + 0.8644 \times 2358.3 = 2289.89$ kJ/kg

Net power produced = $\dot{m}_s\,(h_1 - h_2)$

$= 35\,(3530.9 - 2289.89) = 43435.35$ kW

Thermal efficiency,

Now $h_{f3} = h_{f2}$

Example 4.28

The following data refer to a steam turbine power plant employing one stage of regenerative feed heating:

State of steam entering HP stage: 10 MPa, 600°C

State of steam entering LP stage: 2 MPa, 400°C

State of steam at condenser : 0.01 MPa, $x = 0.9$

The correct amount of steam is bled for feed heating at exit the HP stage. Calculate the mass of steam bled per kg of steam passing through the HP stage and the amount of heat supplied in the boiler per second for an output of 10 MW. Neglect pump work.

[IES, 1993]

Solution

The schematic diagram and *T*-*s* diagram are shown in Fig.4.44(a) and (b).

H.P. stage: $p_1 = 10$ MPa, $t_1 = 600°C$

L.P. stage: $p_2 = 2$ MPa, $t_2 = 400°C$

Condenser: $p_3 = 0.01$ MPa, $x_3 = 0.9$

$W_{out} = 10$ MW

From steam tables, we have (Fig. 4.44(a))

$h_1 = 3625.3$ kJ/kg

$h_2 = 3247.6$ kJ/kg

$h_3 = h_{f3} + x_3\, h_{fg3}$

$= 191.81 + 0.9 \times 2392.8 = 2345.35$ kJ/kg

Heat balance for the heater:

$(1 - m_1)\,(h_5 - h_4) = m_1(h_2 - h_5)$

From steam tables,

$h_5 = 908.77$ kJ/kg at $p_2 = 2$ MPa

$h_4 = 191.81$ kJ/kg at $p_3 = 0.01$ MPa

$\therefore (1 - m_1)(908.77 - 191.81) = m_1(3247.6 - 908.77)$

or $(1 - m_1) \times 716.96 = 2338.83\ m_1$

Figure 4.44 *Regenerative feedwater heating cycle: (a) Schematic diagram, (b) T-s diagram*

or

Heat supplied in boiler,

$q_{\text{in}} = h_1 - h_5 = 3625.3 - 908.8 = 2716.5$ kJ/kg

Work output from boiler,

$$W_{out} = (h_1 - h_2) + (1 - m_1)(h_2 - h_3)$$

$$= (3625.3 - 3247.6) + (1 - 0.235)(3247.6 - 2345.35) = 1067.92 \text{ kJ/kg}$$

Cycle efficiency,

Heat supplied in the boiler for 10 MW output =

Example 4.29

A small power plant produces 25 kg/s steam at 3 MPa, 600°C in the boiler. It cools the condenser with ocean water coming in at 12°C and returned at 15°C. Condenser exit is 45°C. Find (a) the net power output and (b) the required mass flow rate

of ocean water.

[IAS 2012]

Solution

Given that $\dot{m}_s = 25$ kg/s, $p_1 = 3$ MPa, $t_1 = 600°C$, $t_{wi} = 12°C$, $t_{wo} = 15°C$, $T_1 = 600 + 273 = 873$ K, $T_2 = 45 + 273 = 318$ K

The T-s diagram is shown in Fig. 4.45.

1. $h_1 = 3682$ kJ/kg, $h_{f3} = 188.4$ kJ/kg, $p_1 = 9.5934$ kPa, $s_1 = 7.5084$ kJ/kg.K

   $v_{f3} = 0.00101$ m^3/kg, $h_{fg3} = 2394.8$ kJ/kg

   $s_{f3} = 0.6386$ kJ/kg.K, $s_{fg3} = 7.5261$ kJ/kg.K

   Now $s_1 = s_2 = s_{f3} + x_2\, s_{fg3}$

Figure 4.45 *T-s diagram*

   or $7.5084 = 0.6386 + x_2 \times 7.5261$

or $x_2 = 0.9128$

$h_2 = h_{f3} + x_2\, h_{fg3} = 188.42 + 0.9128 \times 2394.8 = 2374.39$ kJ/kg

Pump work, $w_p = v_{f3}\,(p_1 - p_2) = 0.00101\,(300 - 9.59) = 0.29$ kJ/kg

Turbine work, $w_t = h_1 - h_2 = 3682.3 - 2374.39 = 1307.91$ kJ/kg

Net work, $w_{\text{net}} = w_t - w_p = 1307.91 - 0.29 = 1307.62$ kJ/kg

Net power output $= \dot{m}_s \times w_{\text{net}} = 25 \times 1307.62 = 32.69$ MW

2. Heat rejected constant pressure p_2, $q_r = h_2 - h_{f3} = 2374.39 - 188.42 = 2185.97$ kJ/kg

Let $\dot{m}_w$ = mass of ocean water circulated

$$\dot{m}_w c_{pw} \times \Delta t_w = \dot{m}_s \times q_r$$

or $\dot{m}_w \times 4.187 \times (15 - 12) = 25 \times 2185.97$

or $\dot{m}_w = 4350.7$ kg/s

Summary for Quick Revision

1. The Rankine cycle is a vapour pressure cycle which operates on steam and is used for steam power plant. It consists of a

boiler, turbine, condenser, and a pump.

2. Net work, $w_{net} = q_a - q_r = (h_1 - h_{fM}) - (h_2 - h_{f3}) = (h_1 - h_2) - w_p = w_t - w_p$

   Pump work, $w_p = h_{fM} - h_{f3} = v_{f3}\,(p_M - p_3) \times 10^2$ kJ/kg

   $= v_{f3}\,(p_1 - p_3) \times 10^2$ kJ/kg $[\because p_M = p_1]$

   where v_f is in m^3/kg and p is in bar.

3. Thermal efficiency of Rankine cycle considering pump work,

   Heat rate =

4. Thermal efficiency of Rankine cycle neglecting pump work,
5. Specific steam consumption,
6. Overall heat rate, OHR = SSC × heat supplied per kg of throttle steam
7. Back work ratio, and work ratio,
8. Increase of boiler pressure leads to increase in thermal efficiency of ideal Rankine cycle.
9. Decrease in condenser pressure leads to increase in thermal efficiency of ideal Rankine cycle.
10. Thermal efficiency of Rankine cycle can be improved by reheating and regeneration.
11. Efficiency of reheat cycle, η_{reheat}=
12. In the reheat cycle, steam is extracted at a suitable point after expanding in high pressure turbine and is reheated with the help of fl ue gases in the boiler furnace.
13. Regeneration is a method to heat the feed water from the hot well of condenser reversibly by interchange of heat with the system to improve the cycle efficiency
14. In open heaters, the bled steam is allowed to mix with feed water.
15. In closed heaters, the bled steam is not allowed to mix with feed water. The feed water fl ows through the tubes in the heater and bled steam condenses on the outside of tubes.
16. In a binary vapour cycle, two cycles with diff erent working fl uids are coupled in a series. Mercury is used in the topping cycle and steam in the bottoming cycle.

17. In co-generation, some part of the expanded steam in the HP turbine is bled to generate electric power.

Multiple-choice Questions

1. The efficiency of Rankine cycle can be increased by
    1. decreasing initial steam pressure and temperature
    2. increasing exhaust pressure
    3. increasing expansion ratio
    4. increasing regenerative heaters
2. Choose the wrong statement:
    1. bleeding increases the thermodynamic efficiency of turbine
    2. bleeding increases the net power developed by turbine
    3. bleeding decreases the net power developed by turbine
    4. boiler is supplied with hot water due to bleeding.
3. In the Rankine cycle, with the maximum steam temperature being fixed from metallurgical consideration, as the boiler pressure increases,
    1. the condenser load will increase
    2. the quality of turbine exhaust will decrease
    3. the quality of turbine exhaust will increase
    4. the quality of turbine exhaust will remain unchanged
4. Rankine cycle comprises
    1. two isentropic and two reversible constant volume processes
    2. two isentropic and two reversible constant pressure processes
    3. two reversible isothermal and two reversible constant pressure processes
    4. two reversible constant pressure and two reversible constant volume processes.
5. Regeneration effect on Rankine cycle
    1. decreases thermal efficiency
    2. increases thermal efficiency
    3. has no effect on thermal efficiency
    4. increases or decreases thermal efficiency depending on the point extraction of steam.
6. Superheating of steam before expansion in a Rankine cycle
    1. decreases thermal efficiency

2. has no effect on thermal efficiency
3. increases thermal efficiency
4. may increase or decrease thermal efficiency.

7. Reheating of steam
    1. increases thermal efficiency
    2. increases work output of turbine
    3. decreases work output of turbine
    4. increases thermal efficiency but decreases work output.
8. In a binary vapour cycle,
    1. mercury is used in the bottoming cycle
    2. steam is used in the topping cycle
    3. mercury is used in the topping cycle
    4. either mercury or steam may be used in the topping cycle.
9. In a steam power plant, feed water heater is a heat exchanger to preheat feed water by
    1. live steam from steam generator
    2. hot flue gases coming out of the boiler furnace
    3. hot air from air preheater
    4. extracting steam from turbine
10. Consider the following statements regarding effects of reheating of steam in a steam turbine:
    1. It increases the specific output of the turbine.
    2. It decreases cycle efficiency.
    3. It increases blade erosion.
    4. It improves the quality of exit steam.

    Which of these statements are correct?

    1. I and II
    2. II and III
    3. III and IV
    4. I and IV
11. In a steam power plant, what is the outcome of regenerative feed water heating?
    1. Increase in specific output
    2. Increase in cycle efficiency
    3. Improved quality of exhaust steam
    4. Reduced condenser load

    Select the correct answer using the code given

below:

1. I and III only
2. II only
3. II and IV
4. I, II and III

12. In a regenerative feed heating cycle, the economic number of the stages of regeneration
    1. increases as the initial pressure and temperature increase
    2. decreases as the initial pressure and temperature increase
    3. is independent of the intitial pressure and temperature
    4. depends only on the condenser pressure
13. Consider the following statements:

    The reheat cycle helps to reduce

    1. fuel consumption
    2. steam flow
    3. the condenser size

    Which of these statements are correct?

    1. I and II
    2. I and III
    3. II and III
    4. I, II, and III
14. The main advantage of a reheat Rankine cycle is
    1. reduced moisture content in LP side of turbine
    2. increased efficiency
    3. reduced load on condenser
    4. reduced load on pump
15. Consider the following statement pertaining to the features of regenerative steam cycle plant as compared to a non-regenerative plant:
    1. It increases the cycle efficiency.
    2. It requires a bigger boiler.
    3. It requires a smaller condenser.

    Which of the statements are correct?

1. I, II, and III
2. I and II
3. II and III
4. I and III

16. Which of the following statements is not correct for regenerative steam cycle?
    1. It increase thermodynamic efficiency.
    2. It reduces boiler capacity for a given output.
    3. It reduces temperature stresses in the boiler due to hotter feed.
    4. The efficiency increases with increased number of feed heaters.
17. Consider the following for a steam turbine power plant:
    1. Reduction in blade erosion
    2. Increase in turbine speed
    3. Increase in specific output
    4. Increase in cycle efficiency

    Which of these occurs due to reheating of steam?

    1. Only I
    2. I and II
    3. I, III, and IV
    4. II and III
18. When is the greatest economy obtained in a regenerative feed heating cycle?
    1. Steam is extracted from only one suitable point of a steam turbine.
    2. Steam is extracted only from the last stage of steam turbine.
    3. Steam is extracted only from the first stage of a steam turbine.
    4. Steam is extracted from several places in different stages of steam turbines.
19. Consider the following statements:

    The purpose of reheating the steam in a steam turbine power plant is to

    1. increase specific output
    2. increase turbine efficiency
    3. reduce turbine speed

4. reduce specific steam consumption

Which of the statements are correct?

1. II and IV
2. I and III
3. I, II, and IV
4. I, III, and IV

20. In the bottoming cycle of cogeneration, low-grade waste heat is used for
    1. processing
    2. power generation
    3. feed water heating
    4. None of these
21. In steam and other vapour cycles, the process of removing non-condensables is called
    1. scavenging process
    2. deaeration process
    3. exhaust process
    4. condensation process

Review Questions

1. Draw the *T-s* diagram for a simple Rankine cycle and write down the expression for its efficiency.
2. What is the eff ect of boiler pressure and condenser pressure on Rankine cycle efficiency?
3. What are the various methods of improving thermal efficiency of Rankine cycle?
4. Draw the reheat Rankine cycle with superheated steam and write down the expression for its efficiency.
5. Explain regeneration. What are its advantages?
6. What is the diff erence between open heater and closed heater?
7. What are the properties of an ideal working fl uid for the Rankine cycle?
8. What is a binary vapour cycle?
9. What is co-generation?
10. List the advantages of co-generation.
11. What is the diff erence between bleeding and extraction of steam?

4.1 A simple Rankine cycle works between 30 bar and 0.04 bar pressures, the initial condition of steam being dry saturated. Caculate the cycle efficiency, work ratio, and specific steam consumption.

[**Ans.** 35%, 0.997, 3.84 kg/kWh]

4.2 In a Rankine cycle, the steam at inlet to turbine is saturated at pressure of 30 bar and the exhaust pressure is 0.25 bar. Calculate (a) the pump work, (b) the turbine work, (c) the cycle efficiency, and (d) the condenser heat fl ow per kg of steam.

[**Ans.** 3 kJ, 741 kJ, 29.2%, 1790 kJ]

4.3 Steam at 90 bar, 480°C is supplied

to a steam turbine. The steam is reheated to its original temperature by passing the steam through a reheater at 12 bar. The expansion after reheating takes place to condenser pressure of 0.07 bar. Calculate the efficiency of the reheat cycle and work output per kg of steam.

[**Ans.** 42.2%, 1611 kJ/kg]

4.4 A reheat cycle operating between 30 bar and 0.04 bar has a superheat and reheat temperature of 450°C. The first expansion takes place till the steam is dry saturated and then reheat is given. Determine the ideal cycle efficiency neglecting feed pump work.

[**Ans.** 38.7%]

4.5 In a regenerative cycle, the steam pressure at turbine inlet is 30 bar and the

exhaust is at 0.04 bar. The steam is initially saturated. Enough steam is bled off at the optimum pressure to heat the feed water. Neglecting pump work, determine the cycle efficiency.

[**Ans.** 37.3%]

4.6 Steam is supplied to a turbine at pressure of 32 bar, 420°C. It expands isentropically to a pressure of 0.08 bar. Determine the thermal efficiency of the cycle.

If the steam is reheated at 6 bar to 400°C and then expanded isentropically to 0.08 bar, then find the thermal efficiency.

[**Ans.** 35.4%, 36.6%]

4.7 Steam of 28 bar and 50°C superheat

is passed through a turbine and expanded to a pressure where the steam is dry saturated. It is then reheated at constant pressure to its original temperature and then expanded to the condenser pressure of 0.2 bar, the expansion being isentropic. Calculate work done per kg of steam and thermal efficiency with and without reheat.

[**Ans.** 810 kJ/kg, 880 kJ/kg, 30%, 30.3%]

4.8 In a binary vapour cycle, the pressure in the mercury boiler is 15.4 bar and in the mercury condenser is 0.2 bar. The pressure limits for steam cycle are 54 bar and 0.08 bar. If both mercury and steam enter their respective turbines in a dry saturated state, find mass of mercury in kg per kg of steam and

thermal efficiency of the cycle.

[**Ans.** 11.4 kg, 57.61%]

4.9 A binary vapour cycle consists of a mercury operating between 14 bar and 0.1 bar and steam cycle operating between 30 bar, 450°C and 0.04 bar. Calculate the ideal cycle efficiency of the cycle.

[**Ans.** 51.78%]

4.10 Steam is supplied to a regenerative cycle as dry and saturated at 32 bar the condenser pressure is 0.05 bar. If steam is bled at 2.8 bar, find the quality of steam bled, the cycle efficiency, and the specific steam consumption.

[**Ans.** 0.718, 35.27%, 4.342 kg/kWh]

ANSWERS TO MULTIPLE-CHOICE QUESTIONS

1. a
2. c
3. c
4. b
5. b
6. c
7. b
8. c
9. d
10. d
11. c
12. a
13. a
14. a
15. d
16. b
17. d
18. c
19. c
20. b
21. b

Chapter 5

Steam Engines

5.1 ❐ INTRODUCTION

A steam engine uses steam as the working medium. It is the earliest prime mover developed to convert thermal energy into mechanical energy. It is called a prime mover because it is used to drive other devices like locomotives, compressors, lathe, shear presses, etc. On account of their low efficiency steam engines have become obsolete. However, they are still being used in locomotives and some industries.

A steam engine is called an external

combustion engine as the fuel is burnt in the boiler and the steam raised is used to reciprocate the engine piston for generating power.

5.2 ❐ CLASSIFICATION OF STEAM ENGINES

A steam engine is of the reciprocating type, in which the piston moves to-and-fro in the cylinder due to the force applied by the expansion of steam. The reciprocating steam engines may be classified as follows:

1. According to class of service:
    1. Stationary
    2. Marine
    3. Locomotive
    4. Pumping or hoisting
2. According to speed:
    1. **Low speed:** below 125 rpm
    2. **Medium speed:** 125 to 300 rpm
    3. **High speed:** above 300 rpm

    Note that the speed of the engine means speed of the crankshaft.

3. According to arrangement of cylinders:

    1. Horizontal
    2. Inclined
    3. Vertical
4. According to type of valve design:
    1. Simple plate valve
    2. Balance plate valve
    3. Piston valve
    4. Riding cut-off valve
    5. Poppet valve
    6. Corliss valve
5. According to method of governing:
    1. Manual or automatic governing
    2. throttle or cut-off governing
    3. Centrifugal or inertia type governor
6. According to type of exhaust:
    1. Condensing type
    2. Non-condensing type
7. According to steam condition:
    1. High or low pressure
    2. Saturated or superheated
8. According to the action of steam upon the piston:
    1. Single acting
    2. Double acting
9. According to the nature of expansion:
    1. Expansive engines
    2. Non-expansive engines
10. According to the range of expansion of steam:
    1. Simple steam engine
    2. Compound steam engine

5.3 ❐ CONSTRUCTIONAL FEATURES OF A STEAM ENGINE

A schematic line diagram of a vertical steam engine is shown in Fig. 5.1,

The parts of a steam engine may be broadly classified as:

1. Stationary parts
    1. Engine bedplate
    2. Engine frame
    3. Cylinder
    4. Steam chest
    5. Steam jacket
    6. Stuffing box
    7. Cross-head guides
    8. Main bearings
2. Moving parts
    1. Piston
    2. Piston rod
    3. Cross-head

Figure 5.1 *Schematic diagram of a steam engine*

    4. Connecting rod
    5. Crank and crank shaft
    6. D-slide valve
    7. Valve rod
    8. Eccentric and eccentric rod
    9. Flywheel
    10. Governor

5.3.1 Steam Engine Parts

The various parts are briefly described below:

Frame: It is a heavy casting made of cast iron and supports cross-head guides and main bearing. It generally rests on the engine foundation.

Cylinder: It is also a casting to the frame in which the piston reciprocates.

Its both ends are closed and made steam tight. The stuffing box is fixed in one end through which piston rod reciprocates.

Steam chest: It is compartment in a steam engine through which steam is delivered from the boiler to a cylinder. It is connected to the cylinder through the valve passage known as ports. It contains the D-slide valve. The steam is supplied alternately to the cylinder through the ports and exhausted to the condenser. The same port serves as inlet port and exhaust port in double acting engines.

D-slide valve: It connects the cylinder to the steam chest and to the exhaust side through the ports at the correct crank positions. The valve is operated by an eccentric.

Piston: It is made of cast iron. The steam pressure exerts force on it on both sides to reciprocate the piston rod. Piston rings are provided on the piston for steam tightness.

Piston Rod: One end of piston rod is connected to the piston and the other end to the cross-head. It transmits the force from the piston to the cross-head.

Stuffing Box and Gland: It is placed at the point where the piston rod passes through the cylinder cover. It prevents

the leakage of steam from the cylinder to atmosphere, at the same time allowing the piston rod to reciprocate freely.

Cross-Head: It connects the piston rod and connecting rod. It guides the piston rod along a straight line. The thrust on the cross-head is taken of by the cross-head guides.

Connecting Rod: One end of it is connected to the cross-head by a gudgeon pin and the other end is connected to the crank. It transmits force from the cross-head to the crank.It has oscillatory motion.

Crankshaft: The crank converts the reciprocating motion to rotary motion of

the crankshaft.The crank shaft is supported on main bearings and carries the flywheel and eccentric

Eccentric: It converts rotary motion of crankshaft into reciprocating motion of D slide valve.

Valve rod and Eccentric rod: They connect the eccentric to D-slide valve.

Flywheel: It is connected to the crankshaft and helps to maintain a constant angular speed of the engine by storing excess energy during idle stroke of the device being operated.

Governor: It helps in maintaining uniform speed of the engine by regulating supply of stream to the

cylinder.

5.4 ❐ TERMINOLOGY USED IN STEAM ENGINE

A line diagram of steam engine mechanism is shown in Fig. 5.2.

Cylinder Bore, *D*: It is the inside diameter of the cylinder

Piston stroke, *L*: It is the distance travelled by the piston from the cover end of the cylinder to the crank end of cylinder. $L = 2r$.

Single Acting: Steam is supplied on one side of piston only

Double acting: Steam is supplied on both sides of piston.

Dead centres: The position of piston within the cylinder, when the crank and connecting rod are in the same straight line.

Crank radius, *r*: It is the distance between the centre of crankshaft and centre of crank pin

Figure 5.2 *Line diagram of a steam engine*

Swept volume, V_s: It is the volume swept by the piston in one stroke

Mechanical Clearance, L_C: The mechanical clearance is the distance between the cylinder cover and inner dead centre position of the piston. It is specified as percentage of stroke length.

Volumetric Clearance, VC: It is the volume between the cylinder cover and the piston at the inner dead centre position of piston.

Engine speed, N: It is measured in terms of revolutions of crankshaft per minute.

Piston speed: It is linear speed of the piston.

Compression Ratio, α: It is the fraction of stroke volume completed at the start of compression.

Expansion Ratio, **r:** It is the ratio of swept volume to col nine at cut-off of

steam.

Cut off Ratio,: It is the ratio of volume at cut-off steam to the swept volume

Mean Effective Pressure, P_m: It defined as that hypothetical constant pressure which will produce same work for the same piston displacement with the variable conditions of temperature pressure and volume.

Where card area = area of indicator diagram, card length = length of stroke in indicator diagram.

Eccentric Throw: It is the distance between the centre of eccentric and centre of crankshaft.

Valve Travel: It is the maximum distance travelled by the valve along one direction. It is equal to twice the eccentric throw.

Back pressure, p_b: The pressure of exhaust steam acting on the other side of the piston is known as back pressure.

5.5 ❐ WORKING OF A STEAM ENGINE

The working of a double acting steam engine is a shown in Fig. 5 3. As the piston reaches the inner dead centre (*IDC*) position *A*, the *D*-slide valve clears the inlet port C and admits the steam into the cylinder. This high pressure steam pushes the piston on the forward direction and performs work by rotating the crank and eccentric. The steam continues to enter into the

cylinder maintaining a constant pressure inside. With the further rotation of crank and eccentric, *D*-slide valve is moved backwards and closes the inlet port *C*. The steam supply to the cylinder is stopped and this is known as cut-off of steam. After the cut-off the steam in the cylinder expands pushing the piston in the forward direction till the piston reaches the outer dead centre (*ODC*) position *B*. Just before the *ODC* position, the valve connects the cylinder through the port to exhaust *E*. Therefore the pressure on this side of cylinder falls to the atmospheric pressure or condenser pressure. After the *ODC* position the piston reverses its direction of motion and the return stroke starts. The steam is admitted through port *D*

and exhausted on the *IDC* side. Just before the piston reaches the *IDC* position, the port *D* closes and low pressure steam in the cylinder is compressed and acts as a cushion to the piston at the end of the stroke. As the piston reaches the *IDC* position, the D-slide valve opens the port C and the cycle is repeated. When admission and expansion take place on head end side, exhaust and compression takes place on the crank end side.

Figure 5.3 *Working of a steam engine*

5.6 ❐ RANKINE CYCLE

Rankine cycle is the theoretical cycle on which the steam engine works. The Rankine cycle is shown in Fig. 5.4.

The *p*-*v*, *T*-*s* and *h*-*s* diagram for the

Ranking cycle are shown in Fig. 5.5.

Figure 5.4 *Rankine cycle*

Figure 5.5 *Rankine cycle in various ordinates: (a) p-v diagram, (b) T-s diagram, (c) h-s diagram*

The various processes of Rankine cycle are:

Process 1-2: Reversible adiabatic expansion in the engine**.**

Process 2-3: Reversible constant pressure heat transfer in the condenser.

Process 3-4: Reversible adiabatic pumping process in the feed pump.

Process 4-1: Reversible constant pressure heat transfer in the boiler.

The steam on entry into the engine can

be wet, dry saturated or superheated. Considering 1 kg of steam and applying steady flow energy equation to boiler, engine, condenser and pump, we have:

Boiler:

$$h_{f4} + q_1 = h_1$$
$$q_1 = h_1 - h_{f4}$$

Engine:

$$h_1 = w_e + h_2$$

Engine work, $w_e = h_1 - h_2$

Condenser:

$$h_2 = q_2 + h_{f3}$$
$$q_2 = h_2 - h_{f3}$$

Feed pump:

$$h_{f3} + w_p = h_{f4}$$

Pump work, $w_p = h_{f4} - h_{f3}$

Net work, $w_net = w_e - w_p$

Rankine cycle efficiency,

The feed pump handles liquid water which is incompressible. This implies that with the increase in pressure its density or specific volume undergoes a small change. For reversible adiabatic compression, we have

$$T\,ds = ah - v\,dp$$

$$ds = 0$$
$$dh = v\,dp$$

or $\Delta h = v\,\Delta p \qquad [\because \Delta v = 0]$

or $h_{f4} - h_{f3} = v_3\,(p_1 - p_2) \times 10^5$ J/kg

Now. $h_{f4} - h_{f3} = w_p << w_e$, and can be neglected

5.7 ❐ MODIFIED RANKINE CYCLE

In modified Rankine cycle to reduce friction the stroke length is reduced.

Figure 5.6(a) and 5.6(b) show the modified Rankine cycle on *p-v* and T-*s* diagrams respectively, by neglecting pump work.

It may be observed that *p-v* diagram is very narrow at the toe. i.e., point 3′. The

work obtainable near to 3′ is very small which is too inadequate to overcome friction due to reciprocating parts of engine. Therefore the adiabatic process 2-3′ is terminated at point 3. The pressure drop decreases suddenly whilst the volume remains constant. This operation is represented by the line 3-4. By doing so the stroke length is reduced. The loss of work is represented by the hatched area 3-3′-4-3, which is negligibly small.

The work done during the modified Rankine cycle can be calculated as follows:

Let p_1, v_2, u_2, h_2 represents the initial conditions at point 2.

p_2, v_3, u_3, h_3 corresponds to condition of steam at point 3.

Figure 5.6 *Modified Rankine cycle: (a) p-v diagram, (b) T-s diagram*

p_3, h_4 correspond to condition of steam at point 4.

Work done during the cycle per kg of steam = area 1 – 2 – 3 – 4 – 5 – 1

= area 1 – 2 – b – 0 – 1 + area 2 – 3 – a – b – 2 area – 0 – b – a – 4 – 5 – 0

$= p_1 v_2 + (u_2 - u_3) - p_3 v_3$

Heat supplied, $q_s = h_2 - h_{f4}$

5.8 ❒ HYPOTHETICAL OR Theoretical INDICATOR DIAGRAM

The indicator diagram is a representation of the variations of

pressure and volume of steam inside the cylinder on p-V diagram for one complete cycle of operation. The theoretical p-v diagram for a single acting steam engine is shown in Fig. 5.7 without and with clearance.

The sequence of operations is as follows:

Process 1-2: The steam is supplied to cylinder at constant pressure p_1. The steam is cut-off at point 2.

Process 2-3: The steam expands in the cylinder till the piston reaches the ODC position. The expansion process is hyperbolic isothermal, i.e. pv = const.

Process 3-4: The pressure falls from

point 3 to point 4 at constant volume instantaneously due to the opening of the exhaust. The point 3 is called the point of release as the pressure is allowed to fall suddenly to the back pressure p_b.

Process 4-5: It represents the exhaust of used steam at constant pressure

Process 5-1: It represents steam admission to cylinder at constant volume. The pressure suddenly rises form p_b to p_1.

The working cycle is completed within two strokes of piston or one revolution of crank.

Figure 5.7 *Theoretical indicator diagram for a steam engine: (a) Without clearance, (b) With clearance*

Indicator diagram is simply a graph between pressure of steam in a cylinder against the steam volume.

The following assumptions are made in drawing the theoretical indicator diagram:

1. Clearance is neglected.
2. Steam supplied to cylinder is at constant pressure.
3. Ports are opened and closed instantaneously.
4. There is no drop in pressure due to condensation.
5. Expansion of steam in cylinder follows the hyperbolic law.

The theoretical and actual indicator diagrams are shown in Fig. 5.8. The actual indicator diagram is found to be different from the theoretical diagram due to following reason:

1. A definite clearance is necessary to prevent the piston striking the cylinder head. Provide cushioning effect to piston and allow small quantity of water to be collected in the cylinder due to condensation of steam. Therefore the actual supply of

steam starts from point '*a*' instead of point *A*. The admission pressure is also lower due to condensation of Steam on the cooler cylinder walls and throttling effect during flow of steam through the inlet port. Therefore, the supply steam pressure is represented by *ab* instead AB.

2. The instantaneous cut-off is not possible as it takes place during few degrees of crank rotation. Therefore, the diagram gets rounded up near point *b*.
3. The expansion of steam in the cylinder is not perfectly hyperbolic and is represented by some other curve *bc* instead of *BC*. This is due to condensation of steam during expansion.
4. The exhaust opens before the piston reaches *ODC* position (say point C) as instantaneous drop in pressure is not possible. Therefore, fall in pressure starts from C and actual pressure drop is represented by '*cd*'.
5. The exhaust pressure is slightly above the condenser or back pressure as the steam has to be forced out from cylinder. The actual exhaust pressure is represented by '*de*'.
6. The supply of exhaust steam to exhaust port is stopped at point *'c'* by closing the port before the piston reaches *IDC* positions. The entrapped steam is compressed during the remaining stroke and provides a cushioning effect. This is represented by '*ef*'. The inlet port opens just before the piston reaches the *IDC* position and the pressure inside the cylinder increases.

Figure 5.8 *Theoretical and actual indicator diagrams*

The area between the theoretical and actual indicator diagrams represents the lost work. The ratio of the area of the actual indicator diagram to the area of theoretical indicator diagram is called the diagram factor.

5.10 ❐ MEAN EFFECTIVE PRESSURE

The mean effective pressure is a quantity related to the operation of an internal combustion engine. It is a valuable measure of an engine's capacity to do work that is independent of engine displacement.

5.10.1 Without Clearance

The theoretical *p-v* diagram for a single acting steam engine without clearance is shown in Fig. 5.9(a). It can be obtained by drawing a rectangle of area equal to the area of theoretical indicator diagram and base equal to the stroke volume, as shown in Fig 5.9(b).

Area under *p-v* diagram gives the work done per cycle by an engine, Work done per cycle w = (area 1–2–2′–0–1) + (area

2–3 –3′–2′–2) – area (5–4–4′–0–5)

where p_1 = steam admission pressure

p_2 = release or cut-off pressure

Figure 5.9 *p-v diagram without clearance*

p_b = back pressure
r = = expansion ratio
$v_s = v_3$ = swept volume
v_2 = cut-off volume
$v_4 = v_3$, therefore

Now

Work done per cycle as represented by Fig. 5.9(b) is given by

$$w = p_m . v_s$$

or

From the above two equations, we get

5.10.2 With Clearance

The *p-v* diagram for a single acting steam engine with clearance is shown in Fig. 5.10(a). Work done per cycle is given by,

w = area 1-2-3-4-5 -1

=(area 1-2-2′-5′-1) + (area 2-3-3′-2′-2) – (area 5-4-3′-5′-5)

Clearance ratio, $v_3 = v_c + v_s = c\, v_s + v_s =$

$(1 + c)\ v_s$

Expansion ratio,

Cut-off ratio,

Substituting the values of v_c, v_2 and v_3, we get

Figure 5.10 *p-v diagram with clearance*

Mean effective pressure,

5.10.3 With Clearance and Compression

Figure 5.11 shows a hypothetical indicator diagram for a steam engine with clearance and compression. Clearance means the volume of gas left in the cylinder at the discharge end of the stroke. It includes the space between

the piston and cylinder head, the volume of the valves, valve pockets etc. The various processes are:

Process 1**-2**: Steam admission at p_1

Process 2-3: Hyperbolic expansion

Process 3-4: Steam released

Process 4-5: Exhaust of steam into condenser.

Process 5-6: Compression of remaining steam in cylinder

Process 6-1: entry of fresh from boiler into the cylinder rising the cylinder pressure suddenly to boiler pressure p_1.

Work done per cycle w, = area l-2-3-4-5-6-1

$$= (\text{area } 1\text{-}2\text{-}2'\text{-}6'\text{-}1) + (\text{area } 2\text{-}3\text{-}3'\text{-}2'\text{-}2) - (\text{area } 5'\text{-}5\text{-}4\text{-}3'\text{-}5') - (\text{area } 6'\text{-}6\text{-}5\text{-}5'\text{-}6')$$

Figure 5.11 *p-v diagram with clearance and compression*

Let expansion ratio,

Swept volume, $V_s = V_3 - V_c$

Clearance ratio,

$$V_3 = V_c + V_s = cV_s + V_s = (1+c)V_s$$

Cut-off ratio

Compression ratio,

$$V_s = (\alpha + c)V_s$$

5.10.4 With Clearance and Polytropic Expansion and Compression

A polytropic process can be expressed pv_n = constant

The indicator diagram is shown in Fig. 5.12.

Work done per cycle.

w = area l-2-3-4-5-6-1

= (area 1-2-2′-6′-1) + (area 2-3-3′-2′-2) – (area 5-4-3′-5′-5) – (area 6-5-5′-6′-6)

and

Let expansion ratio,

Swept volume, $V_s = V_3 - V_c$

Clearance ratio,

$$V_3 = V_c + V_s = cV_s + V_s = (1+c)V_s$$

Cut-off ratio

Compression ratio,

Figure 5.12 *p-v diagram with clearance and polytopic expansion and compression*

5.11 ❐ POWER DEVELOPED AND EFFICIENCIES

5.11.1 Indicated Power

The power developed by the steam engine based on the indicator diagram is

the indicated power.

Work done by the steam engine per cycle is given by,

w = Force × distance travelled by the piston

Work done per second is the indicated power

where on cover end side.

on crank end side.

d = diameter of piston rod.

If area of piston rod is neglected, then

Let A_i = area of indicator diagram

L_i = length of indicator diagram

The mean height of indicator diagram,

Mean effective pressure, $P_m = h \times s$

where s = spring scale or spring number, $(N/cm^2)/cm$.

5.11.2 Brake Power

A part of the power developed in the engine cylinder is lost in overcoming friction at different parts of the engine. Therefore the power available at the engine crankshaft is less than the indicated power. The power available at the crankshaft is known as the brake power of the engine because it is generally measured by some type of brake.

A rope is wound around the brake drum whose one end is connected to the spring balance suspended from a support and the other end carries the load W, as shown in Fig. 5.13:

Let W = weight on rope, N

S = spring pull, N

D = outer diameter of brake drum

d_r = diameter of rope

The torque applied on brake drum,

Figure 5.13 *Rope brake dynamometer*

5.11.3 Efficiencies of Steam Engine

1. Mechanical efficiency,

Frictional power, $FP = IP - BP$

2. Thermal efficiency =

Heat supplied to engine per kg of steam = $h_1 - h_2$

Where h_1 = heat per kg of steam entering the engine

h_{f2} = heat per kg of condensate coming out of condenser

m_s = mass of steam supplied per hour

Heat supplied per hour = m_s $(h_1 - h_{f2})$

Thermal efficiency of engine on *IP* basis, *i.e.*, indicated thermal efficiency

where m_{si} = specific consumption of steam (kg/kWh) on *IP* basis

where m_{sb} = specific consumption of steam (kg/kWh) on *BP* basis

3. where m_f = mass of fuel burnt per second in the boiler

CV = calorific value of fuel in kJ/kg

5.12 ❐ GOVERNING OF STEAM ENGINES

The automatic variation of steam supply with the variation of load is called governing and the device used for this purpose is known as a governor. There are two methods commonly used for steam engine governing.

1. **Throttle governing:** In this type of governing, the pressure of a steam admitted to the engine is reduced by throttling before it passes into the engine (Fig. 5.14) and the device used for this purpose is called the *throttle valve.* The lower pressure of steam during admission process reduces the work developed by the engine. It has been observed that the steam consumption rate is linearly proportional to the indicated power. This line is called Willams line.
2. **Cut-off governing**: In this system of governing, the period of admission of steam entering the engine cylinder is reduced depending on the load on engine (Fig. 5.14b). The volume of steam admitted is proportional to the mass of steam.

The indicated power *vs* steam consumption per minute for throttle and cut-off governing are shown in Fig. 5.15. It is obvious from the figure that

the steam consumption rate at full load is same for both types of governing, but part load steam consumption of throttle governing is higher than that of cut-off governing engine.

The governor used for throttle governing is simple and cheaper. But the thermal efficiency of cut-off governed engine is higher at part load conditions.

Figure 5.14 *Methods of governing steam engine: (a) Throttle governing, (b) Cut-off governing*

Figure 5.15 *Effect of method of governing on indicated power*

Example 5.1

The steam supply to a steam engine is at 15 bar dry and saturated. The condenser pressure is 0.4 bar.

Calculate the Rankine efficiency of the cycle. Neglect pump work.

Solution

Given: $p_1 = 15$ bar, $p_2 = 0.4$ bar, $x_1 = 1.0$

From steam tables at 15 bar, $h_1 = h_g = 2789.9$ kJ/kg, $s_1 = s_g = 6.4406$ kJ/kg.K

For isentropic expansion of steam,

$$s_1 = s_2 = s_{f2} + x_2\, s_{fg2}$$
$$6.4406 = 1.0261 + x_2 \times 6.6448$$
$$x_2 = 0.815$$
$$h_2 = h_{f2} + x_2\, h_{fg2} = 317.7 + 0.815 \times 2319.2 = 2207.8 \text{ kJ/kg}$$

Example 5.2

A simple Rankine cycle works between 28 bar and 0.06 bar. The initial condition of steam is dry and saturated. Calculate the cycle efficiency, work ratio and specific steam. consumption.

Solution

Given: p_1 = 28 bar, x_1 = 1.0 bar, p_2 = 0.06 bar

The *T-s* diagram for the cycle is shown in Fig. 5.16.

From steam tables:

Figure 5.16

For isentropic process 1-2, we have

$$s_1 = s_2 = s_{f2} = x_2\, s_{fg\,2}$$
$$6.2104 = 0.521 + x_2 \times 7.809$$
$$x_2 = 0.728$$
$$h_2 = h_{f2} + x_2\, h_{fg2} = 151.5 + 0.728 \times 2415.9 = 1910.27 \text{ kJ/kg}$$

Engine work, $w_c = h_1 - h_2 = 2802.0 - 1910.27 = 891.73$ kJ/kg

Pump work, $w_p = h_{f4} - h_{f3} = v_{f3}\,(p_1 - p_2)$

$= 0.001\,(28 - 0.06) \times 10^2 = 2.79$ kJ/kg

Net engine work, $w_{net} = w_c - w_p = 891.73 - 2.79 = 888.94$ kJ/kg

$h_{f4} = h_{f3} + w_p = 151.5 + 2.79 = 154.29$ kJ/kg

Rankine efficiency,

Specific steam consumption,

Example 5.3

A single cylinder double acting engine has a bore of 250 mm and a stroke 300 mm. Steam is admitted at a pressure of 1.3 MPa. Cut-off occurs at 30% of stroke and the exhaust pressure is 0.12 MPa. The diagram factor is 0.82 and the mechanical efficiency is 80%. The engine runs at 150 rpm. Calculate of the power developed.

Solution

Given: $D = 0.5$ m, $L = 0.3$ m, $p_1 =$ 1.3 MPa, , $K_d = 0.82$

$$\eta_{\text{mech}} = 0.8,\ p_b = 0.12\text{MPa},\ N = 150 \text{ rpm}$$

Mean effective pressure,

Brake power developed,

Example 5.4

Dry saturated steam at 0.9 MPa is supplied to a single cylinder double acting steam engine developing 20 kW at 240 rpm. The exhaust is 0.15 MPa. Cut-off takes place at 40% of stroke. The diagram factor is 0.8 and stroke to bore ratio is 1.25. Assuming hyperbolic expansion

determine *(a)* bore and stroke of engine, and (b) steam consumption per hour.

Solution

Given: $p_1 = 0.9$ MPa, $IP = 20$ kW, $N = 240$ rpm, $p_b = 0.15$ MPa

Mean effective pressure,

$$= [0.9 \times 0.4(1 + \ln 2.5) - 0.15] \times 0.8$$
$$= 0.4319 \text{ MPa}$$

Power developed,

Steam consumption/*h*

Specific volume of dry saturated steam at 0.9 MPa = 0.2148 m^3/kg

from steam tables.

Example 5.5

A single cylinder double acting steam engine receives steam at 1.0 MPa and exhaust at 0.055 MPa. The dryness of inlet steam is 96%. The power developed by the engine when running at 220 rpm is 45 kW, with steam consumption of 450kg/h. The engine bore is 0.25 m and stroke is 0.375 m. The expansion ratio is 6. Calculate the diagram factor and the indicated thermal efficiency of the engine.

Solution

Given: $p_1 = 1.0$ MPa, $p_b = 0.055$ MPa, $x = 0.96$, $IP = 45$ kW, $N = 220$ rpm, $m_s = 450$ kg/h, $D = 0.25$ m, $L = 0.375$ m, r = 6

Theoretical m.c.p. p_{mt}

Actual m.e.p $p_m = p_{mt} \times K_d = 0.4103 \times$ Kd

Power developed, IP

Enthalpy of steam at 1 MPa and 0.96 dry, from steam tables,

$$h_1 = h_{f1} + x\, h_{fg} = 762.6 + 0.96 \times 2013.6 = 2717.6 \text{ kJ/kg}$$

Enthalpy of water at exhaust pressure of 0.055 MPa, $h_{f3} = 350.6$

kJ/kg

Indicated thermal efficiency

Example 5.6

A single cylinder double acting condensing type steam engine delivers 20 kW brake power at 240 rpm. The diameter and stroke of the engine are 0.2 m and 0.3 m respectively. The steam is supplied at 10 bar and cut-off takes place at 50% of stroke. The condenser vacuum is 56 cm of Hg while the barometer reads 76 cm of Hg. Mechanical efficiency is 80% clearance is 20 % of stroke and

piston rod diameter is 5 cm. Determine the actual mean effective pressure and diagram factor.

Also determine the specific steam consumption on I.P. basis by neglecting clearance and piston rod area.

Solution

Given: $BP = 20$ kW, $N = 240$ rpm, $D = 0.2$ m, $d = 0.5$ m,

$L = 0.3$ m $p_1 = 10$ bar, , $c = 0.2$, $\eta_{\text{mech}} = 0.8$ $x = 0.95$

Indicated, *m.e.p*

Indicated power,

Actual

Actual *m.e.p.*, bar

Diagram factor,

Neglecting clearance and piston rod area, volume of steam supplied per revolution of engine at 10 bar.

supplied per hour

Steam supplied in kg/h

where v_g = 0.198m3/kg at 10 bar from steam tables

Specific steam consumption on *IP* basis,

Example 5.7

Calculate the diameter and stroke of a single cylinder double acting steam engine developing 50 kW power at 120 rpm with mechanical efficiency of 80%. The steam is supplied at 8 bar pressure and back pressure of the engine is 1.2 bar. Cut-off takes place at 40% of the stroke and clearance is 15% of stroke volume.

Assume diagram factor 0.8, and stroke to bore ratio as 1.5.

Solution

Given: $BP = 560$ kW, $N = 120$ rpm, $\eta_{\text{mech}} = 0.8$, $p_1 = 8$ bar, $p_b = 1.2$ bar,

Indicated, *m.e.p*

$$= 9.8874 \times 102 \times D^3 \text{ kW}$$
$$62.5 = 9.8874 \times 102 \times D^3$$
$$D = 0.398 \text{ m or } 39.8 \text{ cm}$$
$$L = 1.5 \times 0.398 = 59.75 \text{ cm}$$

Example 5.8

The steam enters a steam engine at 15 bar and exhausts at 1.5 bar.

The steam supply is cut-off at 40% of the stroke. The clearance volume is 5% of the swept volume. Calculate the mean effective

pressure.

Solution

Given: $p_1 = 15$ bar, $p_b = 1.5$ bar, $=$ 0.4, $c = 0.05$

Indicated, *m.c.p*

Example 5.9

In a steam engine, the clearance volume is 5% of swept volume and the back pressure is 1.15 bar. If the compression is at 0.3 of the stroke, find the pressure .at the end of compression stroke.

Find also the mean effective

pressure, if the steam supply pressure is 13.7 bar and cut-off occurs at 40% of stroke.

Solution

The *p-v* diagram is shown in Fig. 5.17

Given: $c = 0.05$, $p_1 = 13.7$ bar, $p_b =$ 1.15 bar, = 0.4, $V_5 - V_c = 0.3\ V_s$, a= 0.3 =

Figure 5.17

Example 5.10

The area of indicator diagram 25 cm^2. The swept volume is 0.15 cm^2.

Calculated the theoretical mean effective pressure. The indicator diagram is drawn to the following scales:

1 cm = 1 bar along the pressure axis
1 cm = 0.02 m3 along the volume axis

Spring constant = 1 bar/cm.

Solution

Given: $A_1 = 25\ \text{cm}^2$, V_s = 0.15 cm2,
S = 1 bar/cm

Length of indicator diagram,

M.E.P.,

Example 5.11

Find the diagram factor for steam engine with the following data.

Inlet pressure = 10 bar

Back pressure = 1 bar

Expansion ratio = 3

Area of indicator diagram = 12.1 cm^2

Length of indicator diagram = 7.5 cm

Pressure scale = 3 bar/cm

Solution

Given: $p_1 = 10$ bar, $p_b = 1$ bar, $r = 3$, $A_1 = 12.1\ \text{cm}^2$, $L_1 = 7.5$ cm

Theoretical *m.c.p.*,

Actual *m.e.p.*,

Diagram factor,

Example 5.12

Determine the brake power of a simple double acting steam engine having 400 mm diameter and 500 mm stroke length operating at 350 rpm. The initial and back pressure of steam are 9.5 bar and 1.5 bar respectively. The mechanical

efficiency is 80% and the expansion ratio is 2.5.

Solution

Given: $D = 0.4$ m, $L = 0.5$ m, $N = 350$ rpm, $p_1 = 9.5$ bar, $p_b = 1.5$ bar

Indicated *m.c.p.*,

$$B.P = IP \times \eta_{\text{mech}} = 423.843 \times 0.8 = 339.074 \text{ kW}$$

Example 5.13

A double acting steam engine has a cylinder bore 200 mm stroke 300 mm and cut-off takes place at 0.4 stroke. The steam admission and the

exhaust pressure are 7 bar and 0.38 bar respectively. If the diagram factor is 0.8, calculate the indicated power at 200 rpm. Neglect clearance and assume hyperbolic expansion.

Solution

Given: $D = 0.2$ m, $L = 0.3$ m, $= 0.4$, $p_1 = 7$ bar, $p_b = 0.38$ bar, $K_d = 0.8$, $N = 200$ rpm

Theoretical *m.e.p.*,

Actual *m.e.p.*. $p_{ma} = p_{mt} \times K_d = 4.9856 \times 0.8 = 3.9885$ bar

Power developed,

$= 25.06$ kW

Example 5.14

The following readings were taken during the test at full load on a single cylinder, double acting, condensing type, throttle governed steam engine

Diameter of cylinder = 400 mm

Stroke of engine = 600 mm

Cut-off = 50% of stroke

Pressure of steam supplied = 11 bar

Back pressure = 0.8 bar

Brake wheel diameter = 4.5 m

Net load on the bake = 4900 N

Speed of engine = 150 rpm

Diagram factor = 0.82

Find the indicated power, brake power and mechanical efficiency of the engine.

Solution

Given: $D = 0.4$ m, $L = 0.6$ m, $= 0.5$, $p_1 = 11$ bar, $p_b = 0.8$ bar, $D_b = 4.5$ m, $W_{net} = 4900$ N, $N = 150$ rpm, $K_d = 0.82$

Indicated theoretical *m.e.p.*,

$$= 11 \times 0.5\,(1 + \ln 2) - 0.8 = 8.5123 \text{ bar}$$

Actual *m.e.p.*, $p_{ma} = p_{ml} \times K_d = 8.5123 \times 0.82 = 6.98\text{bar}$

Indicated Power,

Brake power,

Mechanical efficiency,

5.13 ❐ SATURATION CURVE AND MISSING QUANTITY

Saturation Curve: The saturation curve is the curve *showing the volume the steam in the cylinder would occupy, during the expansion stroke if the steam*

is perfectly dry and saturated at all the points. It is plotted on *p*-*V* diagram and the wetness of stream can be seen on it at a glance.

Figure 5.18 shows a calibrated indicator. The total mass of steam in cylinder during expansion stroke = $m_C + m$.

On the expansion curve, consider any point *B* and read off from the diagram the pressure (p_B) and volume (V_B) at this point. From steam tables, obtain the specific volume υ of dry steam at pressure p_B. The volume the steam at *B* would occupy if dry saturated = $m\upsilon$. Let this volume be represented by AC to the volume scale of the *p*-*V* diagram. Then, the point *C* represents the volume the steam at *B* would occupy if *dry*

saturated.

Following this way, a number of points may be obtained and plotted. The curve passing through these points to known as *saturation curve* (because all the points on this line represent the condition of steam dry and saturated).

From this saturation curve, the dryness fraction for all points on expansion curve can be obtained. For instance, the dryness fraction at *E* will be given by

Dryness fraction at

From Fig. 5.18, it can be seen that the steam is wet at the beginning of the expansion stroke and becomes drier towards the end. This is owing to the

fact that high pressure steam in the initial stage of expansion is better than the cylinder walls; this causes the steam to condense. During the expansion stroke the steam pressure falls and towards the end of the stroke the walls will be hotter than the steam; consequently the condensed steam re-evaporates and as a result the dryness fraction is improved. The improvement in dryness fraction will not be there if cylinder walls are not jacketed.

Missing quantity: *The missing quantity is the horizontal distance between the actual expansion curve and the saturation curve* (Fig. 5.18). At a pressure p_D, the missing quantity is represented by *EF* (m^3).

Figure 5.18 *Calibrated indicator diagram*

The missing quantity is mainly due to condensation of the steam, but a small amount will be due to leakage past the piston. Due to this missing quantity there is a loss of work represented by the area between the expansion curve and saturation curve.

The missing quantity can be **reduced** in the following ways

1. By steam jacketing the cylinder walls efficiently.
2. By reducing the temperature range of the steam during the stroke this can be accomplished by *compounding the expansion of steam* in two cylinders instead of allowing the whole pressure to drop to occur in one cylinder.

5.14 ❒ HEAT BALANCE SHEET

Procedure: For preparing a heat balance sheet for a steam engine cylinder, the engine should be tested over a period of time under conditions

of constant load and steam supply. An indicator diagram should be taken and the steam pressure noted at regular intervals of time, an account should also he kept of the steam supply to the jackets. The mass of the steam supplied to the cylinder supplied can be obtained from the steam condensed by the condenser.

Analysis: The heat balance for the steam in engine is more easily drawn up than that of the internal combustion engine. The working fluid in the steam engine does not undergo any chemical change, and consequently changes in properties may be ascertained with reference to an arbitrary datum.

Figure 5.19 shows, diagrammatically, a

steam engine and various quantities entering and leaving have been indicated there on. Treating the engine as a flow system as shown in Fig. 5.20, an energy equation may be written as follows:

where m = mass, W = work, h = specific enthalpy, Q = heat transferred, and suffices have the following meanings, s_1 = steam supplied, s_2 = condensate discharge, c = cooling water, R = radiation.

Figure 5.19 *A steam engine*

Figure 5.20

The Eq. (5.19) in tabular form is expressed as follows:

The value of enthalpy may be obtained from the steam tables. The heat to coding water is given by $m_c \times c_{pw}$ (t_{out} – t_{in}) and the heat to radiation obtained by the difference. In the absence of leakage $m_{s1} = m_{s2}$.

The heat balance may therefore be written as follows:

It is to be noted that friction terms are absent, since any work done against friction is included in heat to the condensate, cooling water and radiation loss.

5.15 ❐ PERFORMANCE CURVES

Typical performance curves of a

reciprocating steam engine under test conditions are shown in Fig. 5.21.

Most of the curves are self-explanatory and should be carefully scrutinised for analysis of the variables involved.

One salient fact is that the speed curve as shown is practically horizontal with a slight drop as the load is increased, this shows good governing. The Willian's line as shown is straight when the governing is by throttling with a fixed cut-off, and indicates a linear relationship between the total steam consumption is kg/h and the power if steam pressure is varied to suit the load. This fact is also of great advantage in predicting the part load steam consumption of steam turbines, most of

which use throttle governing. 1f the steam consumption is plotted for an automatic engine with cut-off governing, the line will not be strictly straight.

Figure 5.21

Example 5.15

In a single cylinder double acting steam engine steam is supplied at a pressure of 12 bar and exhaust takes place at 1.1 bar. The cut off takes place at 10% of the stroke which is equal to 1.25 times the cylinder bore and the engine develops and indicated power of 100 kW at 90 r.p.m., calculated the bore and stroke of the engine assuming

hyperbola expansion and a diagram factor of 0.8. Also determine the theoretical steam consumption in m^3/mm

Solution

Given: p_1 = 12 bar, p_b = 1.1 bar, v_2 = 0.4v_3, L = 1.25D, IP = 100 kW, N = 90 rpm, K_d =0.8

The p-v diagram is shown in Fig. 5.22.

Mean effective pressure without clearance

Expansion ratio,

$$p_m = 0.8\ [12 \times 0.4\ (1 + \ln 2.5) + 1.1] = 6.4786 \text{ bar}$$

Indicated power developed,

Theoretical steam consumption/min

Figure 5.22

Summary for Quick Revision

1. A steam engine is an external combustion heat engine as the fuel is burnt in the boiler. The steam raised is used to reciprocate the piston in the cylinder for generating power.
2. The mean effective pressure is the hypothetical constant pressure which will produce same work for the same piston displacement with the variable conditions of temperature, pressure and volume.

   where

   p_b= back pressure

   p_1= admission pressure of steam

   c = clearance ratio =

3. Efficiency of simple Rankine cycle,

   where, Pump work, $w_p = h_{f4} - h_{f3} = v_{f3}(p_1 - p_2)$

4. Efficiency of modified Rankine cycle
5. The indicator diagram is a representation of the variation of pressure and volume of steam inside the cylinder on *p*-*v* diagram for one complex cycle of operation.
6. Diagram factor is the ratio of the area of actual indicator diagram to the area of theoretical indicator diagram.
7. Indicated power, kW for single acting

   kW for double acting

8. Brake power,

   where D = drop drum dia, d_r = rope dia, N = rpm, W = weight on rope, S = spring pull.

9. Mechanical efficiency,

   Indicated thermal efficiency,

   Brake thermal efficiency,

   Overall efficiency,

10. In throttle governing, the pressure of steam admitted to the engine is reduced by throttling before it passes into the engine.
11. In cut-off governing, the period of admission of steam entering the engine cylinder is reduced depending on the load on engine.
12. Saturation curve is the curve showing the volume the steam in the cylinder would occupy during the expansion stroke if the steam is perfectly dry and saturated at all the points.
13. Missing quantity is the horizontal distance between the actual expansion curve and the saturation curve.
14. Heat balance sheet shows the balance between the amount of heat input to the engine and heat utilised for various purposes.

Multiple-choice Questions

1. Diagram factor is defined as the ratio of

    1. actual m.e.p. and swept volume
    2. theoretical m.e.p. and swept volume
    3. actual m.c.p. and theoretical m.e.p.
    4. theoretical m.e.p. and actual m.e.p.
2. Choose the quantity which is varied during throttle governing of steam engines
    1. pressure
    2. volume
    3. temperature
    4. dryness fraction
3. In a compound steam engine
    1. steam expands twice
    2. two engines are combined together
    3. steam expands in several stages
    4. two units are put together
4. In the tandem compound steam engine, the axes of the two cylinders
    1. are at 90° to each other
    2. are inclined at 45°
    3. are at 0° to each other
    4. lie in different plane
5. In order to reverse a steam engine, the eccentric to be shifted for an angle advance of 40°is
    1. 180°
    2. 200°C
    3. 160°
    4. 190
6. A steam engine works on
    1. Carnot cycle
    2. Rankine cycle
    3. modified Rankine cycle
    4. reheat cycle
7. The cross-head in a steam engine is used to connect
    1. piston rod and connecting rod
    2. valve rod and piston rod
    3. eccentric rod and piston rod
    4. connecting rod and valve rod
8. A D-slide valve in a steam engine is used to
    1. admit steam into cylinder from steam chest.
    2. exhaust steam from cylinder to condenser
    3. to admit and exhaust steam from cylinder
    4. none of the above.
9. The cut-off ratio in a steam engine is defined as the ratio of

volume at cut off to

1. swept volume
2. cylinder volume
3. clearance volume
4. none of above.

10. The clearance ratio in a steam engine is defined as the ratio of clearance volume to
    1. swept volume
    2. cut-off volume
    3. cylinder volume
    4. none of the above
11. Willian's line is the relation between steam mass flow late and
    1. indicated power
    2. brake power
    3. compression ratio
    4. thermal efficiency
12. In receiver type compound steam engine, the cranks are placed at
    1. 0°
    2. 90°
    3. 180°
    4. 270°
13. In a Woolfe compound steam engine, the cranks are placed at
    1. 0°
    2. 90°
    3. 180°
    4. 270°
14. The function of piston in steam engine is to transfer motion in
    1. cross head
    2. connecting rod
    3. D-slide valve
    4. eccentric
15. The function of eccentric in a steam engine is convert
    1. to-and-fro motion of D-slide valve to rotary motion of crankshaft
    2. oscillatory motion of connecting rod to rotary motion of crankshaft
    3. rotary motion of crankshaft to to-end-fro motion of D-slide valve
    4. rotary motion of crankshaft to oscillatory motion of eccentric rod
16. The main function of a stuffing box in a steam engine is to
    1. prevent leakage of steam

2. guide the piston rod
3. guide the valve rod
4. receive exhaust steam from engine

17. The motion to D-slide valve is imparted by
    1. eccentric
    2. cam
    3. flywheel
    4. crank
18. Rankine efficiency of a steam engine may be in the range of
    1. 15–20%
    2. 25–35%
    3. 70–80%
    4. 90–95%
19. Rankine cycle comprises of
    1. two isentropic and two reversible constant volume processes
    2. two isentropic and two reversible constant pressure processes
    3. two isothermal and two reversible constant pressure processes
    4. none of the above

Review Questions

1. Explain the difference between single acting and double acting steam engines
2. Differentiate between condensing and non-condensing steam engines
3. What are the functions of the following
    1. cross head
    2. Eccentric
    3. stuffing box
    4. D-slide valve
4. Explain the following therms:
    1. Cut-off
    2. Release
    3. Dead centres
    4. Back pressure.
5. Define diagram factor.
6. Define indicated thermal efficiency and brake thermal efficiency.
7. List the methods for governing of steam engines.

8. Compare and contrast the methods for governing of steam engine
9. How do you determine the brake power of a steam engine.
10. What is frictional power of a steam engine.

Exercises

5.1 A single cylinder, double acting, condensing type steam engine 0.3 bore and 0.4 m stroke runs at 150 rpm. Steam at 10 bar is supplied upto 40% of the stroke. Back pressure of the engine is 0.35 bar. Neglecting clearance and assuming a diagram factor of 0.8 and a mechanical efficiency of 80% determine the brake power of the engine.

5.2 The following readings were taken from a test on a single cylinder double-acting steam engine having 0.25 m bore and 0.35 m stroke

Pressure of steam supplied = 9.5 bar

Engine speed = 240 rpm

Area of indicator card = 10.5 cm^2

Length of indicator card = 8 cm

Spring number = 25 $N/cm^2/cm$

Net load on brake wheel = 1000 N

Diameter of brake wheel = 3 m

Find *IP*, *BP* and mechanical efficiency

5.3 A single-cylinder, double-acting, non-condensing steam engine 25 cm bore 50 cm stroke develops 40 kW indicated power at 100 rpm. The clearance is 10% and cut-off takes place at 40% of stroke. The steam pressure at

the point of cut-off is 5 bar. The compression starts at 80% of the return stroke. The pressure of steam on compression curve at 90% of the return stroke is 1.4 bar and steam is dry and saturated. Determine the actual and minimum theoretically possible specific steam consumption on indicated power basis.

5.4 Determine the brake power of a single cylinder double acting, condensing type steam engine having 0.3 m bore and 0.6 m stroke, running at 250 rpm. The steam 8 bar dry and saturated is supplied upto 50% stroke. The back pressure is 1 bar. Assume mechanical efficiency 70%, diagram factor = 0.75, and piston rod diameter =

5 cm. Also find the specific steam consumption on indicated power basis if indicated thermal efficiency = 15 %.

5.5 Single cylinder, double acting condensing type steam engine developed 2.10 indicated power at 120 rpm. Dry and saturated steam at 10 bars is supplied for 40% of the stroke. The exhaust pressure is 0.2 bar. Assuming a diagram factor of 0.85 and ratio of stroke to bore as 2 determine the bore and stroke of the engine.

5.6 A single-cylinder, double acting steam engine develops 150 kW indicated power at 240 rpm. The pressure of steam supplied is 10 bar and exhaust pressure is 0.15 bar. The cutoff is at 40 % of the stroke, clearance is 10

% of the stroke and compression starts at 80% of the return stroke. Expansion follows the low: $PV^{1.25} = C$, Compression follow law PV1.35 = C, ratio of stroke to bore diagram factor = 0.85. Calculate the bore and stroke of the engine.

5.7 The following data were obtained from the trail on a steam engine governed by throttle governor steam admission pressure = 10.5 bar dry and saturated

Back pressure = 0.2 bar

Steam consumption at no load = 44 kg/h

Indicated power developed at full load = 7 kW

Specific steam consumption at full load = 18 kg/kWh

Find the specific steam consumption and indicated thermal efficiency when the engine develops 5 kW indicated power.

5.8 A single cylinder, double acting steam engine having 25cm bore and 50 cm stroke is governed by cut-off governing. The maximum cut-off possible is 50 % of stroke. The steam is supplied at 8 bar and exhausts at 1 bar. Calculate the percentage decrease in power when the cut-off is reduce to 40% of stroke. Assume that the diagram factor is 0.8 and mechanical efficiency is 80%.

5.9 A single-cylinder, double-acting,

steam engine develops 40 kW when running at 120 rpm. The steam is admitted at 5 bar and exhausted at 0.35 bar. Cut-off occurs at 1/3 of stroke. Assuming a diagram factor of 0.75 and mechanical efficiency of 0.85 determine the swept volume cylinder.

5.10 Steam is admitted to an engine for 30% of the stroke with pressure of 7 bar, the law of expansion followed is law $pV^{1.5} = C$ the compression commences at 60% of return stoke and follows the law F = C. The clearance volume is 20% of the displacement volume and the back pressure is 6.18 mm of Hg vacuum, when barometer reads 760 mm of Hg. Estimate the mean elective pressure and the indicated power of a

double acting engine with cylinder diameter 30 cm stroke 45 cm and speed 200 rpm.

ANSWERS TO MULTIPLE-CHOICE QUESTIONS

1. c
2. b
3. d
4. c
5. d
6. b
7. d
8. a
9. a
10. a
11. a
12. b
13. c
14. a
15. c
16. a
17. a
18. b
19. b

Chapter 6

Flow Through Steam Nozzles

6.1 ❒ INTRODUCTION

The steam nozzle is a passage of varying cross-section by means of which a part of the enthalpy of steam is converted into kinetic energy as the steam expands from a higher pressure to a lower pressure. Therefore, a nozzle is a device designed to increase the velocity of steam. Steam nozzles are of three types, namely convergent nozzle, divergent nozzle, and convergent–divergent nozzle. If the cross-section of the nozzle decreases continuously from the entrance to exit, then it is called a

convergent nozzle, as shown in Fig. 6.1(a). If the cross-section increases, then it is called a *divergent nozzle*, as shown in Fig. 6.1(b). If the cross-section of the nozzle first decreases and then increases, it is called a *convergent–divergent nozzle*, as shown in Fig. 6.1(c). The least area of cross-section of the nozzle is called the *throat*. The divergent section has to be long as the divergent angle is limited to about 7° in order to prevent separation at the wall.

The main purpose of steam nozzles is to produce a high velocity jet of steam which is used in steam turbine injectors for pumping feed water into boilers and to maintain high vacuum in power plant condensers or steam jet refrigeration

condensers.

Figure 6.1 *Types of nozzles: (a) Convergent nozzle (b) Divergent nozzle (c) Convergent-Divergent nozzle*

6.2 ❐ CONTINUITY EQUATION

Consider the flow of steam through a nozzle.

Let $\dot{m}$ = steady state mass flow rate of steam, kg/s

A = cross-sectional area of nozzle normal to the direction of steam flow at any section, m^2

v = specific volume of steam at the same section, m^3/kg

c = velocity of steam across the section, m/s

Then for steady flow of steam through the nozzle,

Equation (6.1) is the continuity equation for steam flow through the nozzle.

6.3 ❒ VELOCITY OF FLOW OF STEAM THROUGH NOZZLES

The following assumptions are made:

1. The steam flows through the nozzle without any work or heat transfer.
2. The frictional forces are neglected.

Consider the flow of steam through a nozzle, as shown in Fig. 6.2. Applying the energy equation to sections 1 and 2 at entrance and exit of nozzle, we have

Usually $c_1 = 0$,

where $(\Delta h)_{isen} = h_1 - h_2$ = isentropic enthalpy change, kJ/kg

Figure 6.2 *Flow through a nozzle*

6.3.1 Flow of Steam Through the Nozzle

Let pv^n = constant

where $n = 1.035 + 0.1\ x_1$ for wet steam

= 1.135 for saturated steam

= 1.3 for superheated steam

As the steam pressure drops while passing through the nozzle, its enthalpy is reduced. This reduction of enthalpy of steam must be equal to the increase in kinetic energy. Hence,

where p_1, p_2 = pressure of steam at entry and exit, respectively, v_1, v_2 = specific volume at entry and exit, respectively.

If $c_1 << c_2$, then

6.4 ❐ MASS FLOW RATE OF STEAM

The mass flow of steam per second is flowing with velocity c_2 through a cross-sectional area A_2, and specific volume v_2 is:

Now, $v_2 = v_1$

Also, from Eq. (6.3), we have

6.5 ❐ CRITICAL PRESSURE RATIO

From Eq. (6.4), we have

The mass flow rate per unit area is maximum at the throat because it has minimum area of cross-section. Therefore, will be maximum when is maximum

Equation (6.5) gives the critical pressure ratio at which the discharge through the nozzle is maximum.

Substituting for from Eq. (6.5), we get

where c_{s2} = velocity of sound at exit of the nozzle.

where c_{s1} = velocity of sound at entry of

nozzle.

6.6 ❒ MAXIMUM DISCHARGE

From Eq. (6.5), maximum mass flow rate of steam can be obtained as

As p_2 is gradually reduced, the discharge gradually increases and becomes maximum as critical pressure is approached, as shown in Fig. 6.3.

Figure 6.3 *Discharge v's pressure ratio in a nozzle*

6.7 ❒ EFFECT OF FRICTION ON EXPANSION OF STEAM

The exit velocity of steam for a given pressure drop is reduced due to the following reasons:

1. Friction between nozzle surface and steam

2. Internal fluid friction in the steam
3. Shock losses

Most of these losses occur beyond the throat in the divergent section of the nozzle as the length and the velocity of steam is much higher there. The effects of these friction losses are as follows:

1. The expansion does not remain isentropic and the enthalpy drop is reduced, resulting in lower exit velocity.
2. The final dryness fraction of steam is increased as a part of kinetic energy gets converted into heat due to friction and is absorbed by steam, which increases the enthalpy.
3. The specific volume of steam is increased due to frictional reheating.

The effect of friction of steam flow through a nozzle is shown in *h–s* (Mollier) diagram in Fig. 6.4. The point A represents the initial condition of steam and the point E represents the throat of a convergent–divergent nozzle. AB represents the isentropic expansion without friction, and AC represents the

expansion with friction. Isentropic enthalpy drop is $(h_A - h_C)$ and actual enthalpy drop with friction is $(h_A - h_c)$. If the actual enthalpy drop as percentage of theoretical enthalpy drop is known, then point C can be located. However, expansion must end at same pressure as at B. The horizontal line drawn through C to cut the back pressure line p_b at point D represents the final condition of steam. The *h–s* diagram shows that the dryness fraction of steam at point D is greater than that at point B and the specific volume of steam also increases.

Most of the friction occurs in the divergent part of the nozzle and actual expansion is represented by the line AED. AE represents the expansion in

the convergent part, whereas ED represents in the divergent part of the nozzle. Lines AEB and AED represent expansion of steam for initially saturated steam. Likewise, $A_1E_1B_1$ and $A_1E_1D_1$ are for initially superheated steam.

Figure 6.4 *Effect of friction on expansion of steam in a convergent–divergent nozzle*

6.8 ❐ NOZZLE EFFICIENCY

Nozzle efficiency, η_n is a factor that takes into account the effect of friction during expansion of steam in the nozzle. It is defined as:

The exit velocity of steam considering friction is:

where K =

6.9 ❒ SUPERSATURATED OR METASTABLE FLOW THROUGH A NOZZLE

The isentropic expansion of superheated steam from supply pressure p_1 to back pressure p_b can be represented on the Mollier diagram by line AE, as shown in Fig. 6.5. During expansion, change of phase must start at point B where the pressure line p_2 meets the saturation line. However, in nozzles, under certain conditions, this phenomenon of condensation does not occur at point B as the time available is very short due to very high velocity of steam (nearly sonic) through the nozzle. The equilibrium between the vapour phase and liquid is, therefore, delayed and the vapour continues to expand in dry state

even beyond point B upon point B_1. The pressure at B_1 can be found by extending the superheated constant pressure line p_3 up to B_1. The steam during the expansion BB_1 remains dry and condensation is suppressed.

The vapours between pressure p_2 and p_3 are said to be supersaturated or supercooled and such a flow in nozzles is called *supersaturated* or *metastable* flow of steam. A limit to the supersaturated state was observed by Wilson and the line drawn on the Mollier chart through the observed points is known as *Wilson line*. For all practical purposes, this line has become the saturation line.

The flow is also called supercooled flow

because at any pressure between p_2 and p_3, the temperature of the vapour is always lower than the saturation temperature corresponding to that pressure. The difference in this temperature is called the *degree of undercooling*.

When the expansion reaches at point B, on the Wilson line, the condensation occurs at constant enthalpy, and the pressure remaining constant, as shown by horizontal line BC. Further isentropic expansion to the exit pressure is represented by CD. The ratio of saturation pressures corresponding to the temperatures at B and B_1 is called the *degree of supersaturation*.

Figure 6.5 *Super-saturated flow of steam in a nozzle*

Velocity of steam at the end of expansion,

Specific volume, $v_2 = v_1$

Temperature, $T_2 = T_1$

and $A_2 =$

The superheated expansion law, $pv^{1.3} =$ constant, is followed in the supersaturated flow.

Example 6.1

Steam expands from 2.5 bar to 1 bar in a nozzle. The initial velocity of steam is 80 m/s and initial temperature is 200°C. Taking

nozzle efficiency as 96%, find the exit velocity.

Solution

Given that $p_1 = 2.5$ bar, $p_2 = 1$ bar, $c_1 = 80$ m/s, $t_1 = 200°C$, $\eta_n = 0.96$

The $h - s$ diagram is shown in Fig. 6.6.

From steam tables for superheated steam, we have

$h_1 = 2868$ kJ/kg at $p_1 = 2.5$ bar and 200°C

$s_1 = 7.3593$ kJ/kg.K

At $p_2 = 1$ bar, $s_{f2} = 1.3025$ kJ/kg.K

and s_{fg2} = 6.0568 kJ/kg.K

Since $s_1 > s_{fg2}$, steam is superheated. At 1 bar and s_1 = 7.3593 kJ/kg.K, the enthalpy for superheated steam by interpolation is:

Figure 6.6 *Steam flow through a nozzle*

Enthalpy drop = $h_1 - h_2$ = 2868 – 2691.95 = 176.05 kJ/kg

Actual enthalpy drop, $(\Delta h)_{\text{actual}} = \eta_n (h_1 - h_2)$ = 0.96 × 176.05 = 169.0 kJ/kg

Example 6.2

Superheated steam enters a

convergent–divergent nozzle at 20 bar and 300°C. The exit pressure is 4.5 bar. Assuming frictionless flow up to the throat ($pv^{1.3}$ = const.) and a nozzle efficiency of 90%, determine (a) the flow rate for a throat area of 30 cm^2 and (b) exit area.

Solution

Given that $p_1 = 20$ bar, $t_1 = 300$°C, $p_3 = 4.5$ bar, $\eta_n = 0.9$, $A_2 = 30\ cm^2$

Critical pressure ratio, = 0.5457

Throat pressure, $p_2 = 20 \times 0.5457 = 10.91$ bar

From Mollier diagram (Fig. 6.7), we get

$h_1 = 3025$ kJ/kg, $h_2 = 2880$ kJ/kg, $v_2 = 0.2$ m^3/kg, $h_3 = 2700$ kJ/kg, $v_{s3}' = 0.4$ m^3/kg, and $x_3' = 0.995$

Figure 6.7 *Superheated steam flow through a nozzle*

$v_3 = x_{3'}\, v_{s3'} = 0.995 \times 0.4 = 0.398$ m^3/kg

Example 6.3

Dry saturated steam at 3 bar is expanded through a convergent nozzle to 1.5 bar. The exit area is 2.5 cm^2. Calculate the exit velocity and mass flow rate, assuming (a) isentropic expansion, (b) supersaturated flow, and (c) degree of under-cooling at exit.

Solution

1. At $p_1 = 3$ bar

   $s_1 = 6.992$ kJ/kg.K, $h_1 = 2724.3$ kJ/kg, $v_1 = 0.6058$ m^3/kg

   At $p_2 = 1.5$ bar

   $s_1 = s_2 = s_{f2} + x_2\, s_{fg2}$

   $6.9918 = 1.4335 + x_2 \times 5.7897$

   $x_2 = 0.96$

   $h_2 = h_{f2} + x_2 h_{fg2} = 467.08 + 0.96 \times 2226.5 = 2604.52$ kJ/kg

   $v_2 = v_{f2} + x_2\,(v_{g2} - v_{f2}) = 0.001053 + 0.96 \times (1.159 - 0.001053) = 1.113$ m^3/kg

2. For supersaturated flow, $pv^{1.3}$ = const.

3.

   Saturation temperature at 1.5 bar = 111.37°C

   Degree of under cooling at exit = 111.37 – 73.45 = 37.92°C.

Example 6.4

Steam at a pressure of 10 bar, 0.96 dry is expanded through a convergent–divergent nozzle and leaves the nozzle at 0.3 bar.

1. Calculate the velocity of steam at throat for maximum discharge. Take $n = 1.134$.
2. Calculate the exit area and steam discharge if the throat area is 1.5 cm^2. Assume isentropic flow and ignore friction losses.

Solution

The nozzle is shown in Fig. 6.8.

1. $v_1 = v_{f1} + x_1 (v_{g1} - v_{f1}) = 0.001127 + 0.96 \times (0.19444 - 0.001127) = 0.1867$ m^3/kg

   For maximum discharge,

   Velocity of steam at throat, $c_2 =$

Figure 6.8 *Convergent–divergent nozzle*

2. For isentropic flow,

   $s_1 = s_{f1} + x_1 s_{fg1} = 2.1386 + 0.96 \times 4.4478 = 6.4085$ kJ/kg

$s_1 = s_2$

$s_{f1} + x_1 s_{fg1} = s_{f2} + x_2 s_{fg2}$

At $p_2 = 5.774$ bar

$6.4085 = 1.9157 + x_2 \times 4.857$

$x_2 = 0.925$

$h_2 = h_{f2} + x_2 h_{fg2} = 663.92 + 0.925 \times 2091.14$
$= 2598.22$ kJ/kg

$h_1 = h_{f1} + x_1 h_{fg1} = 762.79 + 0.96 \times 2015.3 =$
2697.5 kJ/kg

At $p_3 = 0.8$ bar, $s_{f3} = 1.233$ kJ/kg.K, $s_{fg3} =$
6.202 kJ/kg.K

$s_2 = s_3$

$s_{f2} + x_2\, s_{fg2} = s_{f3} + x_3\, s_{fg3}$

$1.9157 + 0.925 \times 4.857 = 1.233 + x_3 \times 6.202$

$x_3 = 0.833$

$h_3 = h_{f3} + x_3 h_{fg3} = 391.7 + 0.833 \times 2274.1 =$
2286 kJ/kg

Example 6.5

A convergent–divergent nozzle is supplied with steam at 10 bar and 250°C. The divergent portion of the nozzle is 4 cm long and throat diameter is 6 mm. Find the semi-cone angle of the divergent section so that steam may leave the nozzle at 1.2 bar. The frictional loss in the nozzle is 10 percent of the total enthalpy drop. Assume that the frictional loss occurs only in the divergent part of the nozzle.

Solution

Assuming maximum discharge, the throat pressure,

For superheated steam, $n = 1.3$

The nozzle and the $h - s$ (Mollier) diagram is shown in Fig. 6.9. Locate point '1' on the Mollier diagram corresponding to $p_1 = 10$ bar and 250°C. Draw a vertical line from point '1' to cut the p_2 line at point '2' and p_3 line at point 3.

Length (1 – 4) = 0.9 × Length (1 – 3)

Draw horizontal line 4 – 3′ to cut p_3 line at point 3′. Point 3′ gives the exit condition of steam.

From the Mollier diagram, we have

$p_1 = 10$ bar: $h_1 = 2940$ kJ/kg

$p_2 = 5.457$ bar: $h_2 = 2830$ kJ/kg, $v_2 = 0.4$ m^3/kg

$p_3 = 1.2$ bar: $h_3 = 2530$ kJ/kg, $x_{3'} = 0.955$, $v_{s3'} = 1.5$ m^3/kg

$h_1 - h_3 = 2940–2530 = 410$ kJ/kg

$h_4 - h_3 = 0.1 \times 410 = 41$ kJ/kg

Figure 6.9 *Superheated steam flow through a convergent–divergent nozzle*

Example 6.6

Dry saturated steam at pressure of 5

bar flows through a convergent–divergent nozzle at the rate of 4 kg/s and discharges at a pressure of 1.5 bar. The loss due to friction occurs only in the diverging portion of the nozzle and its magnitude is 15% of the total isentropic enthalpy drop. Assume the isentropic index of expansion $n = 1.135$. Determine the area of cross-section at the throat and exit of the nozzle.

Solution

Refer to Fig. 6.10.

For maximum discharge,

From steam table, we get

$h_1 = 2748.7$ kJ/kg, $s_1 = 6.8212$ kJ/kg.K

Now, $s_1 = s_3 = s_{f3} + x_3 s_{fg3}$

$6.8212 = 1.4335 + x_3 \times 5.7997$

$x_3 = 0.93$

$h_3 = h_{f3} + x_3 h_{fg3} = 467.08 + 0.93 \times 2226.5 = 2537.7$ kJ/kg

$(h_1 - h_3)_{\text{isen}} = 2748.7 - 2537.7 = 211$ kJ/kg

$h_1 - h_{3'} = h_1 - h_4 = 0.85 \times 211 = 179.35$ kJ/kg

From Mollier diagram (Fig. 6.10), $x_2 = 0.965$, $x_{3'} = 0.945$, $h_2 = 2662$

kJ/kg

$v_{s2} = 0.7$ m^3/kg, $v_{s3'} = 1.1$ m^3/kg, $h_{3'} = 2585$ kJ/kg

$v_2 = x_2 v_{s2} = 0.965 \times 0.7 = 0.676$ m^3/kg

$v_{3'} = x_{3'} v_{s3'} = 0.945 \times 1.1 = 1.04$ m^3/kg

Figure 6.10 *Dry saturated steam flow through a convergent–divergent nozzle*

6.10 ❒ ISENTROPIC, ONE-DIMENSIONAL STEADY FLOW THROUGH A NOZZLE

A nozzle with both converging and diverging section is shown in Fig. 6.11. For the control volume shown, the following relations can be written:

First law:

Property relation

Continuity equation:

$$\rho A c = \dot{m} = \text{const.}$$

By logarithmic differentiation, we get

Combining Eqs (6.11) and (6.12), we have

Substituting this in Eq. (6.13), we have

Since the flow is isentropic,

and therefore,

Figure 6.11 *One-dimensional isentropic flow through a nozzle*

where c_s = velocity of sound

M = Mach number =

γ = ratio of specific heats = c_p/c_v

This is a very significant equation, and from it, we can draw the following conclusions about the proper shape for nozzles and diffusers:

1. For a nozzle, $dp < 0$. Therefore,

   for a subsonic nozzle, $M < 1$, $dA < 0$, and the nozzle is converging;

   for a supersonic nozzle, $M > 1$, $dA > 0$, and the

nozzle is diverging.

2. For a diffuser, $dp > 0$. Therefore,

for a subsonic diffuser, M < 1, dA > 0, and the diffuser is diverging;

for a supersonic diffuser, M > 1, dA < 0, and the diffuser is converging.

3. When M = 1, dA = 0, which means that some velocity can be achieved only at the throat of a nozzle or diffuser. These conclusions are summarised in Fig. 6.12.

Figure 6.12 *Required area changes for (a) nozzles and (b) diffusers*

6.10.1 Relationship between Actual and Stagnation Properties

The relation between enthalpy h, stagnation enthalpy h_0, and kinetic energy is:

For an ideal gas with constant specific heat, Eq. (6.15) can be written as:

Since = γRT, where $\gamma = c_{p0}/c_{v0}$

For an isentropic process,

Values of are given as a function of M in Table 6.1 for the value of $\gamma = 1.40$.

Table 6.1 *One-dimensional insentropic compressible-flow functions for an ideal gas with constant specific heat and molecular weight and $\gamma = 1.4$*

The conditions at the throat of the nozzle can be found by putting M = 1 at the throat. The properties at the throat are denoted as an asterisk (*) and are referred to as critical properties.

Table 6.2 gives these ratios for various values of *k*.

Table 6.2 *Critical properties for isentropic flow of an ideal gas*

6.11 ❐ MASS RATE OF FLOW THROUGH AN ISENTROPIC NOZZLE

From the continuity equation, we have

Substituting Eq. (6.17) in Eq. (6.22), the flow per unit area can be expressed in terms of stagnation pressure, stagnation temperature, Mach number, and gas properties.

At the throat, M = 1, and therefore, the flow per unit area at the throat, can be found by setting M = 1 in Eq. (6.23).

The area ratio can be obtained by dividing Eq. (6.24) by Eq. (6.23).

Figure 6.13 *Area ratio as a function of Mach number for isentropic nozzle*

The values of A/A* as a function of M are given in Table 6.1. Fig. 6.13 depicts the variation of A/A* with M, which shows that a subsonic nozzle is converging and a supersonic nozzle is diverging.

6.11.1 Effect of Varying the Back Pressure on Mass Flow Rate

Consider first a convergent nozzle as shown in Fig. 6.14, which also shows the pressure ratio p/p_0 along the length of the nozzle. The conditions upstream are the stagnation conditions, which are assumed to be constant. The pressure at

the exit plane of the nozzle is designated p_E and the back pressure (the pressure outside the nozzle exit) p_B. As the back pressure p_B is decreased, the variation of the mass flow rate and the exit plane pressure p_E/p_0 are plotted in Fig. 6.15.

Where $p_B/p_0 = 1$, there is, of course, no flow, and $p_E/p_0 = 1$ as designated by point '*a*'. Next, let the back pressure p_B be lowered to point *b*, so that p_B/p_0 is greater than the critical pressure ratio. The mass flow rate has a certain value and $p_E = p_B$. The exit Mach number is less than one. Next, let p_B be lowered to the critical pressure at point *c*. The Mach number at the exit is now unity and $p_E = p_B$. When p_B is decreased below the critical pressure, designated

by point *d*, there is no further increase in the mass rate of flow, and p_E remains constant at a value equal to the critical pressure, and the exit Mach number is unity. The drop in pressure from p_E to p_B takes place outside the nozzle exit. Under these conditions, the nozzle is said to be choked, which means that for given stagnation conditions, the nozzle is passing the maximum possible mass flow.

Figure 6.14 *Pressure ratio as a function of back pressure for a convergent nozzle*

Figure 6.15 *Mass rate of flow and exit pressure as a function of back pressure for a convergent nozzle*

Figure 6.16 *Nozzle pressure ratio as a function of back pressure for a convergent–divergent nozzle*

Consider next a convergent–divergent nozzle as shown in Fig. 6.16. Point '*a*' designated the condition when $p_B = p_0$

and there is no flow. When p_B is decreased to the pressure indicated by point b, so that p_B/p_0 is less than unity but considerably greater than that the critical pressure ratio, the velocity increases in the convergent section, but M < 1 at the throat. Therefore, the diverging section acts as a subsonic diffuser in which the pressure increases and velocity decreases. Point '*c*' designated the back pressure at which M = 1 at the throat, but the diverging section acts as a subsonic diffuser (with M = 1 at the inlet) in which the pressure increases and velocity decreases. Point '*d*' designates one other back pressure that permits isentropic flow, and in this case, the diverging section acts as a supersonic nozzle with a decrease in

pressure and an increase in velocity. Between the back pressure designated by points *c* and *d*, an isentropic solution is not possible, and shock waves will be present. When the back pressure is decreased below that designated by point *d*, the exit pressure p_E remains constant, and the drop in pressure from p_E to p_B takes place outside the nozzle. This is designated by point *e*.

Example 6.7

A convergent nozzle has an exit area of 500 mm^2. Air enters the nozzle with a stagnation pressure of 1000 kPa and a stagnation temperature of 360 K. Determine the mass rate of flow for back

pressures of 800 kPa, 528 kPa, and 300 kPa, assuming isentropic flow. For air, $k = 1.4$.

Solution

Critical pressure ratio, = 0.528

$$p^* = 1000 \times 0.528 = 528 \text{ kPa}$$

Therefore, for a back pressure of 528 kPa, M = 1 at the nozzle exit and the nozzle is choked.

Decreasing the back pressure below 528 kPa will not increase the flow.

For a back pressure of 528 kPa,

At the exit, $c = c_s = = 347.2$ m/s

Discharge at the exit section,

$$\dot{m} = \rho^* A c = 6.1324 \times 500 \times 10^{-6} \times 347.2 = 1.0646 \text{ kg/s}$$

For a back pressure of 800 kPa, = 0.8

From Table 6.1 for by interpolation

$$T_E = 360 \times 0.9381 = 337.7 \text{ K}$$

Sonic velocity at exit,

Velocity at exit,

Density of air at exit,

$$\dot{m} = \rho_E A_E c_E = 8.2542 \times 500 \times 10^{-6} \times 211.1 = 0.8712 \text{ kg/s}$$

For a back pressure less than the critical pressure (528 kPa), the nozzle is choked and the mass rate of flow is the same as that for the critical pressure. Therefore, for an exhaust pressure of 300 kPa, the mass flow rate is 1.0646 kg/s.

Example 6.8

A converging–diverging nozzle has an exit area to throat area ratio of 2. Air enters this nozzle with a stagnation pressure of 1000 kPa and a stagnation temperature of 360 K. The throat area is 500 mm^2.

Determine the mass rate of flow, exit pressure, exit temperature, exit Mach number, and exit velocity for the following conditions:

1. Sonic velocity at the throat, diverging section acting as a nozzle.
2. Sonic velocity at the throat, diverging section acting as a diffuser.

Solution

1. From Table 6.1, we have

   For = 2, M^*_E = 2.197, = 0.0939, = = 0.5089

   Therefore, $p_E = 0.0939 \times 1000 = 93.9$ kPa

   $T_E = 0.5089 \times 360 = 183.2$ K

   $c_E = M_E^*\, c_{sE} = 2.197 \times 271.3 = 596.1$ m/s

   Critical pressure at throat, $p^* = p_0 \times 0.528 = 1000 \times 0.528 = 528$ kPa

   Critical temperature, $T^* = 0.8333 \times 360 = 300$ K

   At throat,

2. From Table 6.1, we have

For = 2, M_E = 0.308, = 0.936, = 0.9812

Therefore, $p_E = 0.936 \times 1000 = 936$ kPa

$T_E = 0.9812 \times 360 = 353.3$ K

$c_E = MC_{sE} = 0.308 \times 376.8 = 116$ m/s

Since M = 1 at the throat, mass rate of flow is the same as in (a).

6.12 ❒ NORMAL SHOCK IN AN IDEAL GAS FLOWING THROUGH A NOZZLE

A shock wave involves an extremely rapid and abrupt change of state. In a normal shock, this change of state takes place across a plane normal to the direction of flow. Figure 6.17 shows a control surface that includes such a normal shock. Let subscripts *x* and *y* denote the conditions upstream and downstream of shock, respectively, and

assuming steady-state, steady-flow with no heat and work transfer across the control surface, then the various relations are as follows:

First law:

or $h_{ox} = h_{oy}$

Figure 6.17 *One-dimensional normal shock*

Continuity equation:

Momentum equation:

Second law:

The energy and continuity equations can be combined to establish an equation that when plotted on the $h - s$ diagram is

called the *Fanno line*. Similarly, the momentum and continuity equations can be combined to establish an equation the plot of which on the $h - s$ diagram is known as the *Rayleigh line*. Both these lines are shown on the $h - s$ diagram in Fig. 6.18. The point of maximum entropy on each line, points '*a*' and '*b*', corresponds to M = 1. The lower part of each line corresponds to supersonic flow, whereas the upper part corresponds to subsonic flow.

The two points where all three equations are satisfied are points x and y, where x being in the supersonic region and y in the subsonic region. Since $s_y - s_x \geq 0$, the normal shock can proceed only from x to y. This means that the velocity

changes from supersonic (M > 1) before the shock to subsonic (M < 1) after the shock.

Assuming constant specific heats, the energy Eq. (6.26) gives,

That is, there is no change in stagnation temperature across a normal shock. Using Eq. (6.16), we have

and substituting into Eq. (6.30), we have

Figure 6.18 *End states for a one-dimensional normal shock on an enthalpy–entropy diagram*

The continuity equation is,

$$\rho_x c_x = \rho_y c_y$$

But

Combining energy Eq. (6.31) and continuity Eq. (6.32) gives the equation of the *Fanno line*.

The momentum and continuity equations can be combined as follows to give the equation of the *Rayleigh line*.

Combining Eqs (6.33) and (6.34), we get

Table 6.3 gives the normal shock functions, which includes M_y as a function of M_x for $\gamma = 1.4$.

Table 6.3 *One-dimensional normal shock functions for an ideal gas with constant specific heat and molecular weight and $\gamma = 1.4$*

Example 6.9

For the convergent–divergent nozzle of Example 6.8 in which the diverging section acts as a supersonic nozzle (Fig. 6.19), a normal shock stands in the exit plane of the nozzle. Determine the static pressure and temperature and the stagnation pressure (a) just downstream of the normal shock and (b) at a point where M = 1.4.

Solution

1. From Table 6.3, we have

Figure 6.19 *Convergent–divergent nozzle*

$$p_y = 4.46 \times 93.9 = 512.7 \text{ kPa}$$

$T_y = 1.854 \times 183.2 = 339.7$ K

$p_{oy} = 0.630 \times 1000 = 630$ kPa

2. From Table 6.1 at point x, as the flow is isentropic to point x,

Therefore, $p_x = 0.2724 \times 1000 = 272.4$ kPa

$$T_x = 0.6897 \times 360 = 248.3 \text{ K}$$

The properties at y can be determined from the normal shock functions (Table 6.3) as

$p_y = 2.4583 \times 272.4 = 669.6$ kPa

$T_y = 1.320 \times 248.3 = 327.8$ K

$p_{oy} = 0.9298 \times 1000 = 929.8$ kPa

$T_{ox} = T_{oy} = 360$ K, as there is no change in stagnation temperature across a normal shock. From y to E, the diverging section acts as a diffuser.

Example 6.10

Steam at stagnation pressure of 800 $k\text{P}a$ and a stagnation temperature of 350°C expands in a nozzle to 200 $k\text{P}a$. When the mass flow rate is 3 kg/s, determine the throat area and exit area for isentropic flow.

Solution

Critical pressure ratio at the throat, = 0.545

$p^* = 0.545 \times 800 = 436$ kPa

$s^* = s_0 = 7.4089$ kJ/kg. K

$h_0 = 3161.7$ kJ/kg

$T^* = 268.7°\text{C}$

$h^* = 3001.4$ kJ/kg

At the nozzle exit,

$p_E = 200$ kPa, $s_E = s_0 = 7.4089$ kJ/kg.K

$T_E = 178.5°C$

$h_E = 2826.7$ kJ/kg

$v_E = 1.0284$ m^3/kg

Example 6.11

5 kg/s of steam at 30 bar and 350°C is supplied to a group of 6 nozzles

in a wheel diameter maintained at 4 bar. Calculate the following:

1. The dimensions of the nozzles of rectangular cross-sectional flow area with aspect ratio of 2.5:1. The expansion may be considered metastable and friction is neglected.
2. Degree of undercooling and supersaturation
3. Loss in available heat drop due to irreversibility
4. Increase in entropy
5. Ratio of mass flow rate with metastable expansion to that if expansion is in thermal equilibrium.

Solution

The *h*-*s* diagram is shown in Fig. 6.20.

$$h_1 = 3115.3 \text{ kJ/kg}$$

$$v_1 = 0.09053 \text{ m}^3\text{/kg}$$

$$s_1 = 6.74257 \text{ kJ/kg.K}$$

$$s_3 = s_1 = s_{f3} + x_3 s_{fg3}$$

$$6.7427 = 1.7766 + x_3 \times 5.1193$$

$$x_3 = 0.97$$

$$h_3 = h_{f3} + x_3 h_{fg3}$$

$$= 604.73 + 0.97 \times 2133.8$$

$$= 2674.5 \text{ kJ/kg}$$

$$v_3 = v_{f3} + x_3(v_{g3} - v_{f3}) = 0.0010840.97 \times (0.4625 - 0.001084)$$

$$= 0.45 \text{ m}^3\text{/kg}$$

Figure 6.20 *Mollier diagram for superheated steam*

1. For supersaturated steam, $n = 1.3$

   Let breadth $= b$

   Then length, $l = 2.5\ b$

   Area of 6 nozzles $= 6 \times 2.5b \times b = 15b^2$

$15b^2 = 2.279 \times 10^{-3}$

$b = 0.0123$ m or 12.3 mm

$l = 0.0308$ m or 30.8 mm

2. At 4 bar, $t_s = 143.65°C$

   Degree of undercooling = 143.65 – 118.34 = 24.31°C

   p_s corresponding to 118.3°C = 1.9 bar

   Degree of super saturation = = 2.1

3. $h_1 = h_3 = 3115.3 - 2674.5 = 440.8$ kJ/kg

   Loss of available heat drop = $(h_1 - h_3) - (h_1 - h_2)$

   $(\Delta h)_{loss} = 440.8 - 437.5 = 3.3$ kJ/kg

4. Increase in entropy
5.

Example 6.12

Dry saturated steam at 5 bar with negligible velocity expands

isentropically in a convergent nozzle to 1 bar and dryness fraction 0.94. Determine the velocity of steam leaving the nozzle.

Solution

Given that $p_1 = 5$ bar; $p_2 = 1$ bar; $x_2 = 0.94$

For $p_1 = 5$ bar, enthalpy of dry saturated steam from steam tables,

$$h_1 = h_{g1} = 2748.7 \text{ kJ/kg}$$

and for $p_2 = 1$ bar, we find that,

$h_{f2} = 417.44$ kJ/kg, $h_{fg2} = 2258.0$ kJ/kg

$$\therefore h_2 = h_{f2} + x_2\, h_{fg2}$$

$= 417.44 + 0.94 \times 2258.0 = 2539.96$ kJ/kg

Enthalpy drop, $\Delta h = h_1 - h_2$

$= 2748.75 - 2539.96 = 208.74$ kJ/kg

Velocity of steam leaving the nozzle

Example 6.13

Dry saturated steam at a pressure of 15 bar enters in a nozzle and is discharged at a pressure of 1.5 bar. Find the final velocity of steam when initial velocity of steam is

negligible. If 15% of the heat drop is lost in friction, then find the percentage reduction in the final velocity.

Solution

For $p_1 = 15$ bar from steam tables,

$h_1 = 2792.1$ kJ/kg

and for $p_2 = 1.5$ bar,

$h_2 = 2693.5$ kJ/kg

Enthalpy drop, $\Delta h = h_1 - h_2$

$= 2792.19 - 2693.5 = 98.6$ kJ/kg

Final velocity of steam,

Percentage reduction in the final enthalpy = 15% = 0.15

Nozzle coefficient or nozzle efficiency, $\eta_n = 1 - 0.15 = 0.85$

Final velocity of steam,

Percentage reduction in final velocity

Example 6.14

Steam approaches a nozzle with velocity of 250 m/s at a pressure of 3.5 bar and dryness fraction 0.95. If the isentropic expansion in the nozzle proceeds till the pressure at

exit is 2 bar, then determine the change of enthalpy and dryness fraction of steam. Calculate also the exit velocity from the nozzle and the area of exit of nozzle for flow of 0.75 kg/s.

Solution

Given that $c_1 = 250$ m/s, $p_1 = 3.5$ bar, $x_1 = 0.95$, $p_2 = 2$ bar, $\dot{m} = 0.75$ kg/s

From Mollier diagram, as shown in Fig. 6.21,

Figure 6.21 *Mollier diagram*

$h_1 = 2624$ kJ/kg

$h_2 = 2534$ kJ/kg

$\Delta h = h_1 - h_2 = 2624 - 2534 = 90$ kJ/kg

Dryness fraction, $x_2 = 0.92$

From steam tables, $v_{f2} = 0.001061$ m^3/kg, $v_{g2} = 0.8857$ m^3/kg at $p_2 = 2$ bar

Specific volume at exit

$$v_2 = v_{f2} + x_2 (v_{g2} - v_{f2}) = 0.001061 + 0.92 \times (0.8857 - 0.001061) = 0.8149 \text{ m}^3/\text{kg}$$

Now, $= h_1 - h_2$

Steam mass flow rate ($\dot{m}$) is given by,

Example 6.15

Dry saturated steam at a pressure of 6 bar flows through a convergent–divergent nozzle at a rate of 4.5 kg/s and discharges at a pressure of 1.6 bar. The loss due to friction occurs only in the diverging portion of nozzle and its magnitude is 12% of total isentropic enthalpy drop. Assuming the isentropic index of expansion, $n = 1.135$, determine the cross-sectional area at throat and exit of nozzles.

Solution

Given that $p_1 = 6$ bar, $\dot{m} = 4.5$ kg/s, $p_3 = 1.6$ bar, $n = 1.135$, $\eta_n = 1 - 0.12 = 0.88$

From Mollier diagram shown in Fig. 6.22,

$h_1 = 2750$ kJ/kg, $h_3 = 2520$ kJ/kg

Refer to $h - s$ diagram (Mollier chart) as shown in Fig. 6.22.

Let suffix 1, 2 and 3 denote entrance, throat, and exit.

We have

$p_2 = 6 \times 0.578 = 3.468$ bar and

$h_2 = 2650$ kJ/kg

Figure 6.22 *Mollier diagram*

For Section 1–2:

$$h_1 - h_2 = 2750 - 2650 = 100 \text{ kJ/kg}$$
$$A_2 = \text{Area of throat}$$
$$v_2 = 0.964 \text{ m}^3\text{/kg}$$

From Mollier diagram, we have

$$v_{g2} = 0.5321 \text{ m3/kg corresponding to } p_2 = 3.468 \text{ bar}$$
$$v_2 = x_2\, v_{g2} = 0.964 \times 0.5321 = 0.513 \text{ m}^3\text{/kg}$$

For Section 2–3:

$$(h_1 - h_3) \text{ isentropic} = 230 \text{ kJ/kg}$$
$$h_1 - h_{3'} = (h_1 - h_3) \times \eta_n = 230 \times 0.88 = 202.4 \text{ kJ/kg}$$

From Mollier diagram,

$x_{3'} = 0.9365$

$v_{g3} = 1.0911$ m^3/kg corresponding to pressure of 1.6 bar.

$v_3 = x_{3'}\, v_{g3} = 0.9365 \times 1.0911 =$

1.022 m^3/kg

Example 6.16

Dry saturated steam enters a nozzle at a pressure of 10 bar and velocity of 100 m/s. The discharge pressure is 5 bar and discharge velocity is 500 m/s. Heat loss from the nozzle is 5 kJ/kg. Determine the final dryness fraction of steam.

Solution

Given that p_1 = 10 bar, c_1 = 100 m/s, p_2 = 5 bar, c_2 = 500 m/s, q = −5

kJ/kg

Using steady flow energy equation

Since $w = 0$

At $p_1 = 10$ bar, $x_1 = 1.0$, $h_1 = 2778.1$ kJ/kg

At 5 bar; $h_{f2} = 640.21$ kJ/kg, $h_{fg2} = 2108.5$ kJ/kg

Example 6.17

Steam enters a group of nozzles of a steam turbine at 12 bar and 220°C and leaves at 1.2 bar. The steam

turbine develops 220 kW with a specific steam consumption of 13.5 kg/kWh. If the diameter of nozzles at throat is 7 mm, then calculate the number of nozzles.

Solution

Given that $p_1 = 12$ bar, $t_1 = 220°C$, $p_3 = 1.2$ bar, power developed = 220 KW, $m_3 = 13.5$ kg/kWh, and

$d_2 = 7$ mm

Refer to Mollier diagram, as shown in Fig. 6.23.

The steam is initially superheated.

$n = 1.3$ and $p_2/p_1 = 0.5457$

$p_2 = 0.5457\ p_1 = 0.5457 \times 12 = 6.548$ bar

From Mollier diagram, we have

$h_1 = 2860$ kJ/kg, $h_2 = 2750$ kJ/kg, $x_2 = 0.992$

Figure 6.23 *Mollier diagram*

$v_{g2} = 0.29$ m^3/kg

$\Delta h_2 = h_1 - h_2 = 2860 - 2750 = 110$ kJ/kg

A_2 = Area of throat

Mass flow rate per nozzle

Total mass flow rate,

Example 6.18

Dry saturated steam at a pressure of 8 bar enters a convergent–divergent nozzle and leaves it at a pressure of 1.5 bar. If the flow is isentropic and the corresponding expansion index is 1.135, then find the ratio of cross-sectional area at exit and throat for maximum discharge.

Solution

Given that $p_1 = 8$ bar, $p_3 = 1.5$ bar, $n = 1.135$

Let A_2 = Area of cross-section at throat

A_3 = Area of cross-section at exit

m = mass of steam discharged per second

Refer to Mollier diagram as shown in Fig. 6.24.

Figure 6.24 *Mollier diagram*

For dry saturated steam, $n = 1.135$

From Mollier chart, $h_1 = 2775$ kJ/kg

$$h_2 = 2650 \text{ kJ/kg}$$
$$h_3 = 2465 \text{ kJ/kg}$$
$$x_2 = 0.965,\ x_3 = 0.902$$

From steam tables, $v_{g2} = 0.405$ m3/kg, $v'_{g2} = 1.159$ m3/kg

Ratio of cross-sectional area at exit and throat =

Example 6.19

Steam at a pressure of 10 bar and dryness fraction of 0.98 is discharged through convergent–divergent nozzle to a back pressure of 0.1 bar. The mass flow rate is 10 kg/kWh. If the power developed is 220 kW, then determine the following:

1. Pressure at throat
2. Dimensions at exit of the nozzle if the throat is of rectangular cross-section of 5 mm × 10 mm.

The loss due to friction is 10% of the overall isentropic enthalpy drop in the divergent portion.

Solution

Refer to the Mollier diagram as shown in Fig. 6.25.

1. The isentropic index of expansion for wet steam:

   $n = 1.035 + 0.1x_1$

   $n = 1.035 + 0.1 \times 0.98 = 1.133$

   Now $p_1 = 10$ bar,

   $p_2 = 10 \times 0.5778 = 5.778$ bar

2. From Mollier diagram, we have

   $h_1 = 2762$ kJ/kg, $h_3 = 2075$ kJ/kg, $h_2 = 2660$ kJ/kg

   Loss in heat drop due to friction = $0.1\,(h_1 - h_3)$ = $0.1\,(2762 - 2075) = 68.7$ kJ/kg

$$h_{3'} - h_3 = 68.7$$
$$h_{3'} = h_3 + 68.7 = 2075 + 68.7 = 2143.7 \text{ kJ/kg}$$

Figure 6.25 *Mollier diagram*

Mass flow rate,

Velocity at throat,

Total area of throat,

$$x_2 = 0.953$$
$$v_{g2} = 0.34 \text{ m}^3/\text{kg at 5.778 bar (from steam tables)}$$

Throat area per nozzle = $5 \times 10 \times 10^{-6} = 0.5 \times 10^{-4}$ m^2

Number of nozzles = = 7.97 = 8 (say)

Exit area/nozzle = = 7.476×10^{-4} m^2

Keeping the same aspect ratio of 1:2 for rectangle and let x be the smaller side

$$2x \times x = 7.476 \times 10^{-4}$$
$$2x^2 = 7.476 \text{ cm}^2$$

Therefore, exit rectangle is 19.33 mm × 38.66 mm.

Example 6.20

The dry saturated steam is expanded in a nozzle from pressure of 10 bar to pressure of 5 bar. If the expansion

is supersaturated, then find (a) the degree of under cooling, and (b) the degree of supersaturation.

Solution

Given that p_1 = 10 bar and p_2 = 5 bar

From steam tables corresponding to p_1 = 10 bar, saturation temperature of steam.

T_1 = 179.91°C = 452.91 K

$T_{2'}$ = Temperature at which supersaturation occurs

From steam tables corresponding to a pressure of 5 bar, we find

saturation temperature

$$T_2 = 151.86°\text{C}$$

∴ Degree of undercooling

$$= T_2 - T_{2'} = 151.86 - 112.88 = 38.98°\text{C}$$

Example 6.21

Steam enters a nozzle in a dry saturated condition and expands from a pressure of 2 bar to 1 bar. It is observed that supersaturated flow is taking place and steam flow reverts to normal flow at 1 bar. What is the degree of undercooling and increase in entropy and loss in available heat drop due to irreversibility?

Solution

For supersaturated flow,

From steam tables at 2 bar,

$$T_1 = (120.23 + 273) = 393.23 \text{ K}$$

Hence, = 335 K

From steam tables corresponding to 1 bar, T_2 (saturation temperature)

$$= 99.62°\text{C} = 99.62 + 273 = 372.62 \text{ K}$$

Degree of undercooling

$$= T_2 - T_{2'} = 372.62 - 335 = 37.62 \text{ K}$$

Refer to the Mollier diagram as shown in Fig. 6.26.

Corresponding to 335 K, saturation

pressure from steam tables = 0.22 bar.

$\therefore p_{2'} = 0.22$ bar

Figure 6.26 *Mollier diagram*

Degree of supersaturation = = 4.545

For supersaturated flow, the formula for superheated steam is used.

Thus,

where $n = 1.3$

$v_1 = 0.8857$ m^3/kg from steam tables

Isentropic heat drop = $(h_1 - h_2)$ = 117.5 kJ/kg (from Mollier diagram)

Therefore, loss in availability =
117.5 – 113.5 = 4 kJ/kg

Increase in entropy = = 0.01073 kJ/kg.K

Example 6.22

Find the percentage increase in discharge from a convergent–divergent nozzle expanding steam from 8.75 bar dry to 2 bar, when (a) the expansion is taking place under thermal equilibrium and (b) the steam is in metastable state during part of its expansion.

Take area of nozzle as 2500 mm^2.

Solution

Given that $p_1 = 8.75$ bar, $p_2 = 2$ bar, $A_2 = 2500\ \text{mm}^2 = 2500 \times 10^{-6}\ \text{m}^2$

Refer to the Mollier diagram shown in Fig. 6.27.

1. Mass of steam discharged when expansion is under thermal equilibrium.

   Let $\dot{m}_1$ = Mass of steam discharged

   From Mollier diagram, $h_1 = 2770$ kJ/kg, $h_2 = 2515$ kJ/kg, and $x_2 = 0.91$

   From steam tables,

   Figure 6.27 *Mollier diagram*

   $$v_{g2} = 0.885\ \text{m}^3/\text{kg}$$
   $$\Delta h_2 = h_1 - h_2 = 2770 - 2515 = 255\ \text{kJ/kg}$$

   Velocity of steam at exit,

2. Mass of steam discharged when it is in metastable state.

   Let $\dot{m}_2$ = mass of steam discharged

   Volume of steam at inlet,

Volume of steam at exit,

Furthermore,

Heat drop from inlet to exit

$$\Delta h_2 = h_1 - h_2 = 2770 - 2530 = 240 \text{ kJ/kg}$$

Velocity of steam at exit,

∴ Percentage increase in discharge

Example 6.23

Steam expands through a nozzle from 5 bar and dry saturated to a back pressure of 0.2 bar. Mass flow is 2 kg/s. Calculate the exit and the throat areas under the following conditions:

1. Isentropic expansion with negligible velocity

2. Isentropic expansion with initial velocity of 100 m/s
3. Friction loss at pressure amounts to 10% of the total heat drop up to that pressure and initial velocity negligible

Solution

1. Pressure at throat is given by

$$p_2 = 0.5774 \times p_1 = 0.5774 \times 5 = 2.887 \text{ bar}$$

Drop a vertical from point 1 to cut pressure for 2.887 bar at point 2 and the pressure line for 0.2 bar, i.e., the back pressure at point 3. Then from the Mollier diagram (Fig. 6.28), the following values are obtained:

At point 1, $h_1 = 2745$ kJ/kg

At point 2, $p_2 = 2.887$ bar, $h_2 = 2645$ kJ/kg, and $v_2 = 0.63$ m^3/kg

At point 3, $p_3 = 0.2$ bar, $h_3 = 2248$ kJ/kg, and $v_3 = 6.6$ m^3/kg

Figure 6.28 *Mollier diagram*

Hence, the area at throat for a mass flow of 2 kg/s is given by,

Similarly, area at exit for a mass flow of 2 kg/s is given by,

2. Refer Fig. 6.29.

Hence, $h_0 = 5.0 + h_1 = 5 + 2745 = 2750$ kJ/kg

Corresponding to $h_0 = 2750$ kJ/kg

$p_0 = 5.1$ bar as shown on the chart

Thus, the problem is resolved to the condition of $p_0 = 5.1$ bar and $h_0 = 2750$ kJ and velocity of zero at entry to the nozzle.

Therefore, pressure at throat is given by,

And at $p_2 = 2.945$ bar

$h_2 = 2653$ kJ/kg, $v_2 = 0.61$ m^3/kg, from the Mollier chart

$p_3 = 0.2$ bar

$h_3 = 2248$ kJ/kg, $v_3 = 6.6$ m^3/kg, from the Mollier chart

Figure 6.29 *Mollier diagram*

Area at throat is given by,

Similarly, area at the exit from the nozzle is given by,

The nozzle efficiency $\eta_n = 0.9$ and $n = 1.135$

for dry saturated steam.

Therefore, m = = 1.119

The critical pressure ratio is given by,

And the discharge per unit area is given by,

Area at throat, A_2 =

Using the same formula and putting the value of r_p = we get

Therefore, area at exit is given by,

The area at the exit may also be found with the help of Mollier diagram (Fig. 6.30). Isentropic enthalpy drop as found in part (a) = 497 kJ/kg.

Therefore, the actual heat drop = 0.9 × (2745 – 2248) kJ/kg

= 497 + 0.9 = 447.3 kJ/kg

Figure 6.30 *Mollier diagram*

i.e., $h_1 - h'_3$ = 447.8 kJ/kg

and v_3' = 6.7 m^3/kg

Therefore, area at exit is given by,

Example 6.24

A convergent–divergent nozzle is required to discharge 2 kg of steam per second. The nozzle is supplied with steam at 7 bar and 180°C and discharge takes place against a back pressure of 1 bar. The expansion up to throat is isentropic and the frictional resistance between the throat and the exit is equivalent to 63 kJ/kg of steam. Taking approach velocity of 75 m/s and throat pressure of 4 bar, estimate: (a) suitable areas for the throat and exit and (b) overall efficiency of the nozzle based on the enthalpy drop between the actual inlet pressure

and temperature and the exit pressure.

Solution

Given that $\dot{m}$= 2 kg/s; p_1 = 7 bar, T_1 = 180°C, p_3 = 1 bar, frictional resistance = 63 kJ/kg of steam, c_1 = 75 m/s, p_2 = 4 bar

1. Suitable areas for the throat and exit

   Let A_2 = area at the throat

   A_3 = area at the exit

   The expansion of steam through the nozzle on the Mollier diagram is shown in Fig. 6.31. From the Mollier diagram, we find that

   Figure 6.31 *Mollier diagram*

   h_1 = 2810 kJ/kg, h_2 = 2680 kJ/kg, h_3 = 2470 kJ/kg, x_2 = 0.97, and x_3 = 0.934

   From steam tables, we also find that the specific volume of steam at throat corresponding to 4 bar,

$v_{f2} = 0.001084$ m^3/kg, $v_{g2} = 0.4625$ m^3/kg

$v_2 = v_{f2} + x_2(v_{g2} - v_{f2}) = 0.001084 + 0.97(0.4625 - 0.001084) = 0.449$ m^3/kg

and specific volume of steam corresponding to 1 bar,

$v_{f3} = 0.001043$ m^3/kg, $v_{g3} = 1.694$ m^3/kg

$v_3 = v_{f3} + x_3(v_{g3} - v_{f3}) = 0.001043 + 0.934(1.694 - 0.001043) = 1.582$ m^3/kg

Heat drop between entrance and throat,

$\Delta h_2 = h_1 - h_2 = 2810 - 2680 = 130$ kJ/kg

∴ Velocity of steam at throat,

Since there is a frictional resistance of 63 kJ/kg of steam between the throat and the exit,

$h_3 - h'_3 = 63$ or $h_3 = h'_3 + 63 = 2470 + 63 = 2533$ kJ/kg

and heat drop between entrance and exit,

$\Delta h_3 = h_1 - h_3 = 2810 - 2533 = 277$ kJ/kg

∴ Velocity of steam at exit,

2. Overall efficiency of the nozzle

Overall efficiency of the nozzle

Example 6.25

Air is expanded reversibly and adiabatically in a nozzle from 13 bar and 150°C to a pressure of 6 bar. The inlet velocity of the nozzle is very small and the process occurs under steady-state flow conditions. Calculate the exit velocity of the nozzle.

[IES, 1992]

Solution

Given that $p_1 = 13$ bar, $T_1 = 273 + 180 = 423$ K, $p_2 = 6$ bar, and $c_1 = 0$

SFEE for unit mass flow rate is:

Figure 6.32 *Flow through steam nozzle*

Since the air expands reversibly and adiabatically from state 1 to 2, as shown in Fig. 6.32, we have $q = 0$, $w = 0$, and $z_1 = z_2$

Exit velocity of the nozzle,

Example 6.26

A steam nozzle receives steam at 40 bar and 400°C at an initial velocity of 40 m/s. The final pressure of steam is 10 bar. The mass flow rate of steam is 2 kg/s. The nozzle efficiency is 90%. The cross-section

of the nozzle is circular. The angle of divergence is 6°. Calculate the throat and exit diameters and the length of the divergent portion. Show the representation of process on $h - s$ diagram and sketch the nozzle and label the dimensions calculated.

[IES, 2005]

Figure 6.33 *Flow through convergent-divergent steam nozzle*

Solution

The convergent–divergent nozzle is shown in Fig. 6.33 and $h - s$ diagram in Fig. 6.34.

From steam tables and Mollier diagram, we have

h_1 =3213.5 kJ/kg for p_1 = 40 bar and t = 400°C

p_2 = 0.5457, p_1 = 0.5457 × 40 = 21.828 bar, for superheated steam

h_2 = 3025 kJ/kg, v_{s2} = 0.125 m^3/kg

$c_2 = [2(h_1 - h_2) \times 10^3 +]^{0.5}$

$= [2(3213.5 - 3025) \times 10^3 + 40^2]^{0.5}$
= 615.3 m/s

Figure 6.34 *h − s diagram*

In Fig. 6.34,

Summary for Quick Revision

1. A steam nozzle may be defined as a passage of varying cross-section by means of which a part of enthalpy of steam is converted into kinetic energy as the steam expands from a

higher pressure to a lower pressure.

2. Continuity equation: $\dot{m}$
3. Velocity of steam through a nozzle:

For

where h_1 and h_2 are in kJ/kg

4. Isentropic flow through a nozzle
   1. pv_n= constant

      where $n = 1.035 + 0.1\, x_1$ from wet steam

      = 1.135 for saturated steam

      = 1.3 for superheated steam

   2. Velocity, c_2 =
   3. Mass of steam discharged,
   4. Critical pressure ratio, for maximum discharge.
   5. Velocity at exit for maximum discharge,
   6. Maximum discharge, $(\dot{m})_{max}$ = A_2
5. Most of the friction losses occur beyond the throat in the divergent section of the nozzle.
6. Nozzle efficiency,
7. Exit velocity considering friction in nozzle,
8. Supersaturated (or meta stable) flow:

In steam nozzles, the condensation of steam beyond the saturation line is suppressed or delayed due to very high velocity of steam and the vapours continue to expand in dry state even beyond the saturation line. The steam vapours are said to be supersaturated or supercooled in this region.

Velocity of steam at the end of expansion,

9. Wilson line represents the limit to the supersaturated state on the Mollier chart.
10. Isentropic, one-dimensional steady flow through a nozzle:

    1. where

    2. For a nozzle, $dp < 0$.

       Subsonic nozzle: $M < 1$, $d\mathrm{A} < 0$, and nozzle is converging.

       Supersonic nozzle: $M > 1$, $d\mathrm{A} > 0$, and nozzle is diverging.

    3. For a diffuser, $dp > 0$.

       Subsonic diffuser, $M < 1$, $d\mathrm{A} > 0$, and diffuser is diverging.

       Supersonic diffuser, $M > 1$, $d\mathrm{A} < 0$, and diffuser is converging.

    4. When $M = 1$, $d\mathrm{A} = 0$, which occurs at the throat of nozzle and diffuser.
11. Stagnation properties:
12. Properties at the throat, $M = 1$
13. Mass rate of flow through isentropic nozzle.

    At the throat, $M = 1$

14. Area ratio,
15. A shock wave involves an extremely rapid and abrupt change of state.
    1. In a normal shock, the abrupt change of state takes place across a plane normal to the direction of flow.

2. The normal shock proceeds from supersonic to subsonic region.
3. There is no change in stagnation temperature across a normal shock, i.e., $T_{ox} = T_{oy}$.
4.
5.
6.

Multiple-choice Questions

1. A nozzle has velocity head at outlet of 10 m. If it is kept vertical, the height reached by the stream is
    1. 100 m
    2. 10 m
    3.
    4.
2. The effect of friction on flow of steam through a nozzle is to
    1. decrease the mass flow rate and to increase the wetness at the exit
    2. increase the mass flow rate and to increase the exit temperature
    3. decrease the mass flow rate and to decrease the wetness of the steam
    4. increase the exit temperature, without any effect on the mass flow rate
3. For adiabatic expansion with friction through a nozzle, the following remains constant
    1. Entropy
    2. Static enthalpy
    3. Stagnation enthalpy
    4. Stagnation pressure
4. The critical pressure ratios for flow of dry saturated steam and superheated steam through a nozzle are, respectively,
    1. 0.5279 and 0.550
    2. 0.577 and 0.550
    3. 0.577 and 0.546
    4. 0.5279 and 0.546
5. At which location of a converging–diverging nozzle, does the shock-boundary layer interaction take place?
    1. Converging portion
    2. Throat
    3. Inlet

4. Diverging portion

6. The effect of friction in a steam nozzle is to
   1. increase velocity and increase dryness fraction
   2. increase velocity and decrease dryness fraction
   3. decrease velocity and increase dryness fraction
   4. decrease velocity and decrease dryness fraction
7. Consider the following statements:

   When dry saturated or slightly superheated steam expands through a nozzle,

   1. The coefficient of discharge is greater than unity.
   2. It is dry up to Wilson's line.
   3. Expansion is isentropic throughout.

   Of these statements

   1. I, II, and III are correct
   2. I and II are correct
   3. I and III are correct
   4. II and III are correct
8. The total and static pressures at the inlet of a steam nozzle are 186 kPa and 178 kPa, respectively. If the total pressure at the exit is 180 kPa and static pressure is 100 kPa, then the loss of energy per unit mass in the nozzle will be
   1. 78 kPa
   2. 8 kPa
   3. 6 kPa
   4. 2 kPa
9. Under ideal conditions, the velocity of steam at the outlet of a nozzle for a heat drop of 400 kJ/kg will be approximately
   1. 1200 m/s
   2. 900 m/s
   3. 60 m/s
   4. the same as the sonic velocity
10. Consider the following statements:
    1. De Laval nozzle is a subsonic nozzle.
    2. Supersonic nozzle is a converging passage
    3. Subsonic diffuser is a diverging passage

    Which of these statements is/are correct?

1. I and II
2. II and III
3. I alone
4. III alone

11. In a steam nozzle, inlet pressure of superheated steam is 10 bar. The exit pressure is decreased from 3 bar to 1 bar. The discharge rate will
    1. remain constant
    2. decrease
    3. increase slightly
    4. increase or decrease depending on whether the nozzle is convergent or convergent–divergent
12. Air from a reservoir is to be passed through a supersonic nozzle so that the jet will have a Mach number of 2. If the static temperature of the jet is not to be less than 27°C, the minimum temperature of air in the reservoir should be
    1. 48.6°C
    2. 167°C
    3. 267°C
    4. 367°C
13. Consider the following statements:

    For supersaturated flow through a steam nozzle, the

    1. Enthalpy drop reduces further
    2. Exit temperature increases
    3. Flow rate increases

    Which of these statements are correct?

    1. I, II, and III
    2. I and II
    3. II and III
    4. I and III
14. The critical pressure ratio for maximum discharge through a nozzle is given by
    1.
    2.
    3.
    4.
15. Consider the following statements in relation to a convergent–

divergent steam nozzle operating under choked conditions:

1. In the convergent portion, steam velocity is less than sonic velocity.
2. In the convergent portion, steam velocity is greater than sonic velocity.
3. In the divergent portion, the steam velocity is less than sonic velocity.
4. In the divergent portion, the steam velocity is greater than sonic velocity.

Which of the above statements are correct?

1. I and III
2. I and IV
3. II and III
4. II and IV

16. For maximum discharge through a convergent nozzle, the pressure ratio should be (where *n* is the index for expansion)
    1.
    2.
    3.
    4.
17. Wilson line is associated with which one of the following?
    1. Total steam consumption with respect to power output
    2. Supersonic flow of steam through a nozzle
    3. Nozzle flow with friction
    4. Supersaturated flow of steam through a nozzle
18. In a steam nozzle, to increase the velocity of steam above sonic velocity by expanding steam below critical pressure
    1. a vacuum pump is added
    2. ring diffusers are used
    3. divergent portion of the nozzle is necessary
    4. abrupt change in cross-section is needed
19. Which one of the following is the correct statement? To get supersonic velocity of steam at nozzle exit with a large pressure drop across it, the duct must
    1. converge from inlet to exit
    2. diverge from inlet to exit
    3. first converge to the throat and then diverge till exit
    4. remain constant in cross-section
20. Which one of the following is the correct expression for the critical pressure ratio of a nozzle?

1.
2.
3.
4.

21. What is the critical pressure ratio for isentropic nozzle flow with ratio of specific heats as 1.5?
    1. $(0.8)^3$
    2. $(0.8)^{0.6}$
    3. $(1.25)^{0.33}$
    4. $(1.25)^3$
22. A compressible fluid flows through a passage as shown in Fig. 6.35. The velocity of the fluid at the point A is 400 m/s.

Figure 6.35

Which one of the following is correct?

At point B, the fluid experiences

    1. an increase in velocity and decrease in pressure
    2. a decrease in velocity and increase in pressure
    3. a decrease in velocity and pressure
    4. an increase in velocity and pressure
23. If the cross-section of a nozzle is increasing in the direction of flow in supersonic flow, then in the downstream direction
    1. both pressure and velocity will increase
    2. both pressure and velocity will decrease
    3. pressure and increase but velocity will decrease
    4. pressure will decrease but velocity will increase
24. If the velocity of propagation of small disturbances in air at 27°C is 330 m/s, then at a temperature of 54°C, its speed would be
    1. 660 m/s
    2. 300 × m/s
    3. 330/ m/s
    4. 330 × m/s
25. For one-dimensional isentropic flow in a diverging passage, if the initial static pressure is p_1 and the initial Mach number is M_1 ($M_1 < 1$), then for the downstream flow
    1. $M_2 < M_1$; $p_1 < p_2$
    2. $M_2 < M_1$; $p_2 > p_1$
    3. $M_2 > M_1$; $p_2 > p_1$
    4. $M_2 > M_1$; $p_2 < p_1$

26. Which of the following diagrams correctly depict the behaviour of compressible fluid flow in the given geometries (Fig. 6.36)?

Figure 6.36

Codes:

1. 1 and 4
2. 2 and 4
3. 2 and 3
4. 1 and 3

27. Figure 6.37 represents a schematic view of the arrangement of supersonic wind tunnel section. A normal shock can exist without affecting the test conditions

Figure 6.37

1. Between Sections 4 and 5
2. At Section 4
3. Between Sections 4 and 3
4. Between Sections 1 and 2

28. The Mach number for nitrogen flowing at 195 m/s when the pressure and temperature in the undisturbed flow are 690 kN/m abs and 93°C, respectively, will be
    1. 0.25
    2. 0.50
    3. 0.66
    4. 0.75
29. The Mach number at inlet of gas turbine diffuser is 0.3. The shape of the diffuser would be
    1. converging
    2. diverging
    3. stagnation enthalpy
    4. stagnation pressure
30. The identical convergent–divergent nozzles A and B are connected in series, as shown in Fig. 6.38 to carry a compressible fluid.

Figure 6.38

Which one of the following statements regarding the velocities at the throats of the nozzles is

correct?

1. Sonic and supersonic velocities exist at the throats of the nozzles A and B, respectively.
2. Sonic velocity can exist at throats of both nozzles A and B.
3. Sonic velocity will always exist at the throat of nozzle A, whereas subsonic velocity will exist at throat of nozzle B.
4. Sonic velocity exists at the throat of nozzle B, whereas subsonic velocity exists at the throat of nozzle A.

31. Semi-angle of a Mach cone is
    1.
    2. $\sin^{-1} M$
    3.
    4.

where M is the Mach number.

32. Which of the following statements are correct?
    1. Mach wave is a very weak shock wave.
    2. Entropy change across a shock wave is nearly zero.
    3. Total pressure behind a shock wave is less than that ahead of it.
    4. Mach number behind a normal shock is less than one.

Select the correct answer using the codes given below:

Codes:

1. I, II, and III
2. I, III, and IV
3. I, II, and IV
4. II, III, and IV

33. Introduction of a Pitot tube in a supersonic flow would produce
    1. normal shock at the tube nose
    2. curved shock at the tube nose
    3. normal shock at the upstream of the tube nose
    4. curved shock at the upstream of the tube nose

34. In the Fanno line shown in Fig. 6.39,

Figure 6.39

1. subsonic flow proceeds along PQR
2. supersonic flow proceeds along PQR
3. subsonic flow proceeds along PQ and supersonic flow proceeds along RQ
4. subsonic flow proceeds along PQ and supersonic flow proceeds along QR

35. Given k = ratio of specific heats, for Rayleigh line, the temperature is maximum at a Mach number of

1.
2.
3. $1/k$
4. k

36. What is/are the effect(s) of supersaturation in nozzle flow?

1. It increases the mass flow
2. It increases friction in the nozzle
3. It increases exit velocity
4. It reduces dryness fraction of steam.

Select the correct answer using the code given below:

1. I only
2. I and III
3. I, II and IV
4. III only

37. Consider the following statements pertaining to isentropic flow:

1. To obtain stagnation enthalpy, the flow need not be decelerated isentropically but should be decelerated adiabatically.
2. The effect of friction in an adiabatic flow is to reduce the stagnation pressure and increase entropy.
3. A constant area tube with rough surfaces can be used as a subsonic nozzle.

Of these statements:

1. I, II, and III are correct
2. I and II are correct

3. I and III are correct
4. II and III are correct

38. Consider the following statements:

A convergent–divergent nozzle is said to be choked when:

1. Critical pressure is attained at the throat.
2. Velocity at throat becomes sonic.
3. Exit velocity becomes supersonic.

Of these statements

1. I, II, and III are correct
2. I and II are correct
3. II and III correct
4. I and III are correct

39. In flow through a convergent nozzle, the ratio of back pressure to the inlet pressure is given by the relation

If the back pressure is lower than p_B given in the above equation, then:

1. The flow in the nozzle is supersonic
2. A shock wave exists inside the nozzle
3. The gases expand outside the nozzle and a shock wave appears outside the nozzle
4. A shock wave appears at the nozzle exit

40. Consider the following statements:

Across the normal shock, the fluid properties change in such a manner that the:

1. Velocity of flow is subsonic
2. Pressure increases
3. Specific volume decreases
4. Temperature decreases

Of these statements:

1. II, III, and IV are correct
2. I, II, and IV are correct
3. I, III, and IV are correct
4. I, II, and III are correct

41. Match List I (Property ratios as the critical and stagnation conditions) with List II (values of ratios) and select the correct answer using the codes given below the lists:

Codes:

	A	B	C	D
1.	2	1	4	4
2.	1	2	3	4
3.	2	1	3	4
4.	1	2	4	3

42. For oblique shock, the downstream Mach number
    1. is always more than unity
    2. is always less than unity
    3. may be less or more than unity
    4. can never be unity
43. Fanno line flow is a flow in a constant area duct
    1. with friction and heat transfer but in the absence of work
    2. with friction and heat transfer and accompanied by work
    3. with friction but in the absence of heat transfer and work
    4. without friction but accompanied by heat transfer and work
44. Reyleigh line flow is a flow in a constant area duct
    1. with friction but without heat transfer
    2. without friction but with heat transfer
    3. with both friction and heat transfer
    4. without either friction or heat transfer
45. The plot of the pressure ratio along the length of the convergent–divergent nozzle is shown in Fig. 6.40. The sequence of the flow conditions labelled (1), (2), (3), and (4) in

the figure is, respectively

Figure 6.40

1. Supersonic, sonic, subsonic, and supersonic
2. Sonic, supersonic, subsonic, and supersonic
3. Subsonic, supersonic, sonic, and subsonic
4. Subsonic, sonic, supersonic, and subsonic

46. Acoustic velocity in an elastic gaseous medium is proportional to
    1. absolute temperature
    2. stagnation temperature
    3. square root of absolute temperature
    4. square root of stagnation temperature
47. Which of the following statement(s) is/are relevant to critical flow through a steam nozzle?
    1. Flow rate through the nozzle is minimum.
    2. Flow rate through the nozzle is maximum.
    3. Velocity at the throat is supersonic.
    4. Velocity at the throat is sonic.

Select the correct answer using the codes given below:

Codes:

1. I alone
2. I and III
3. II and IV
4. IV alone

48. Which of the following statements is/are true in case of one-dimensional flow of perfect gas through a converging–diverging nozzle?
    1. The exit velocity is always supersonic.
    2. The exit velocity can be subsonic or supersonic.
    3. If the flow is isentropic, then the exit velocity must be supersonic.
    4. If the exit velocity is supersonic, then the flow must be isentropic.

Select the correct answer from the codes given below:

Codes:

1. II and IV
2. II, III, and IV
3. I, III, and IV
4. II alone

49. In a normal shock, in a gas
    1. the stagnation pressure remains the same on both sides of the shock
    2. the stagnation density remains the same on both sides of the shock
    3. the stagnation temperature remains the same on both sides of the shock
    4. the Mach number remains the same on both sides of the shock
50. A normal shock
    1. causes a disruption and reversal of flow pattern
    2. may occur only in a diverging passage
    3. is more severe than an oblique shock
    4. moves with a velocity equal to the sonic velocity
51. In a normal shock, in a gas
    1. both the downstream flow and the upstream flow are supersonic
    2. only the upstream flow is supersonic
    3. the downstream flow is sonic
    4. the upstream flow is subsonic
52. Which of the following parameters decrease across a normal shock wave?
    1. Mach number
    2. Static pressure
    3. Stagnation pressure
    4. Static temperature

Select the correct answer using the code given below:

1. Only I and III
2. Only II and IV
3. I, II, and III
4. II, III, and IV

53. When are shock waves formed in air compressors?
    1. Mach number < 0.9

2. Mach number > 0.9
3. Mach number = 2
4. Mach number changes suddenly from one value to another

54. It is recommended that the diffuser angle should be kept less than 18° because
    1. pressure decrease in flow direction and flow separation may occur
    2. pressure decreases in flow direction and flow may become turbulent
    3. pressure increases in flow direction and flow separation may occur
    4. pressure increases in flow direction and flow may become turbulent
55. A converging diverging nozzle is connected to a gas pipeline. At the inlet of the nozzle (converging section) the Mach number is 2. It is observed that there is a shock in the diverging section. What is the value of the Mach number at the throat?
    1. < 1
    2. Equal to 1
    3. > 1
    4. ≥ 1
56. A nozzle is discharging steam through critical pressure ratio. When the back pressure is further decreased, the nozzle flow rate will
    1. decrease
    2. increase
    3. remain unaltered
    4. first increase to a maximum and then will decrease
57. In isentropic flow between two points
    1. the stagnation pressure decreases in the direction of flow
    2. the stagnation temperature and stagnation pressure decrease with increase in the velocity
    3. the stagnation temperature and stagnation pressure may vary
    4. the stagnation temperature and stagnation pressure remain constant
58. Steam flows at the rate of 10 kg/s through a supersonic nozzle. Throat diameter is 50 mm. Density ratio and velocity ratio with reference to throat and exit are respectively 2.45 and 0.8. What is the diameter at the exit?
    1. 122.5 mm

2. 58 mm
3. 70 mm
4. 62.5 mm

59. Which of the following is caused by the occurrence of a normal shock in the diverging section of a convergent-divergent nozzle?
    1. Velocity jump
    2. Pressure jump
    3. Velocity drop
    4. Pressure drop

    Select the correct answer using the codes given below:

    1. I only
    2. I and II
    3. II and III
    4. I and IV
60. Total enthalpy of stream at the inlet of a nozzle is 2800 kJ while static enthalpy at the exit is 2555 kJ. What is the steam velocity at the exit if expansion is isentropic?
    1. 70 m/s
    2. 245 m/s
    3. 450 m/s
    4. 700 m/s
61. What is/are the effect(s) of supersaturation in nozzle flow?
    1. It increases the mass flow
    2. It increases friction in the nozzle
    3. It increases exit velocity
    4. It reduces dryness fraction of steam.

    Select the correct answer using the code given below:

    1. I only
    2. I and III
    3. I, II, and IV
    4. III only

Explanatory Notes

1. 1. (b) Velocity head = = h = 10 cm
2. 8. (c) Pressure loss = $p_{oi} - p_{oe}$ = 186 – 180 = 6 kPa
3. 9. (b) Velocity of steam at outlet of nozzle = = 894.4 ≃ 900 m/s

1. 11. (a) Critical pressure ratio for superheated steam = 0.5457

$p_2 = 10 \times 0.5457 = 5.457$ bar

Exit pressure of 3 bar is less than p_2. With further decrease of pressure from 3 bar to 1 bar, discharge rate will remain the same.

2. 12. (a)

$= 1 + 0.5\,(1.4 - 1) \times 2^2 = 1.8$

$T_r = 27 \times 1.8 = 48.6$°C

3. 21. (a) Critical pressure ratio =
4. 24. (d)
5. 28. (b) For nitrogen,

Velocity of sound in nitrogen,

6. 36. (d) Flight Mach number of Aircraft = = 2
7. 58. (c) $A_t\, c_t\, \rho_t = A_e\, c_e\, \rho_e$

$= 0.8 \times 2.45 \times 50^2 = 4900$

$d_e = 70$ mm

8. 60. (d) c_e = = 700 m/s

Review Questions

1. Define a nozzle.

2. Differentiate between a nozzle and a diffuser.
3. What is a throat of nozzle?
4. What is the value of n in pv^n = constant for saturated steam?
5. Write the formula for critical pressure ratio in terms of index of expansion.
6. Define Mach number.
7. What are the reasons for reduction in velocity of steam in the nozzle?
8. What are the effects of friction losses in the nozzle?
9. Define nozzle efficiency.
10. What is metastable steam flow?
11. What is Wilson line?
12. Write the conditions for a nozzle to be subsonic and supersonic.
13. Define stagnation enthalpy and stagnation pressure.
14. Define back pressure.
15. What is a normal shock?
16. Explain Fanno line and Rayleigh line.

Exercises

6.1 Steam is expanded in a set of nozzles from 10 bar and 200°C to 5 bar. What type of nozzle is it? Neglecting the initial velocity find minimum area of the nozzle required to allow a flow of 3 kg/s under the given conditions. Assume isentropic expansion of steam.

[**Ans.** Convergent, 21 cm^2]

6.2 Steam expands in a nozzle from 4 bar and 200°C to 1 bar. If the velocity of steam at entry is 60 m/s and nozzle efficiency is 92%, determine the exit velocity.

[**Ans.** 687.7 m/s]

6.3 Dry saturated steam enters a convergent–divergent nozzle at 11 bar and exit at a pressure of 2 bar. The flow is frictionless and isentropic, following the law $pv^{1.135}$ = constant. Calculate (a) the exit velocity of steam and (b) the ratio of cross-section areas at exit and throat.

[**Ans.** 774.6 m/s, 1.62]

6.4 A convergent–divergent nozzle is to be designed in which steam at 14 bar and 275°C is to be expanded to 1.05 bar.

Calculate the necessary throat and exit diameter of the nozzle for steam discharge of 500 kg/h. Assume that 12% of the total heat drop is lost due to friction only in the divergent part of the nozzle.

[**Ans.** 9.9 mm, 17.2 mm]

6.5 Steam at 15 bar, 350°C enters a nozzle of the convergent–divergent type and leaves at 1 bar. The throat diameter is 6 mm and length of the diverging part is 80 mm. Determine the cone angle of the divergent part. Assume that 12 per cent of the total available enthalpy drop is lost in friction in the divergent part only.

[**Ans.** 4° 22′]

6.6 A convergent–divergent nozzle is

required to discharge 2 kg/s of stream. The nozzle is supplied with steam at 6.9 bar, 180°C and discharge takes place against a back pressure of 0.98 bar. Expansion up to throat is isentropic and the frictional resistance between the throat and the exit is equivalent to 62.76 kJ/kg of steam. Taking approach velocity of 75 m/s and throat pressure 3.9 bar, estimate: (a) suitable areas for the throat and exit and (b) overall efficiency of the nozzle based on the enthalpy drop between the actual inlet pressure, temperature and the exit pressure.

[**Ans.** 17.66 cm^2, 41.67 cm^2, 82.25%]

6.7 The nozzles in the stage of an impulse turbine receive steam at 17.5

bar, 300°C and pressure in the wheel chamber is 10.5 bar. If there are 16 nozzles, then find the cross-sectional area at the exit of each nozzle for a total discharge to be 280 kg/min. Assume nozzle efficiency of 90%.

[**Ans.** 1.36 cm^2]

6.8 Steam at a pressure of 10 bar and 95% dry is supplied through a convergent–divergent nozzle to the wheel chamber where the pressure is maintained at 0.12 bar. The mass flow rate through the nozzle is 8.16 kg/kWh and the work developed by the wheel is 110 kW. Determine the following:

1. Pressure of the throat for maximum discharge
2. Number of nozzles required if throat diameter of each nozzle is 5 mm
3. The exit diameter of nozzle if 10% of overall enthalpy drop overcomes friction in the divergent portion

[**Ans.** 5.93 bar, 9, 18.1 mm]

6.9 Five kilogram of steam per minute passes through a convergent–divergent nozzle. The pressure and temperature of steam supplied to the nozzle is 10 bar and 200°C, respectively. The discharge pressure is 0.1 bar. The expansion is supersaturated up to throat and in thermal equilibrium afterwards. Calculate (a) the area of nozzle at exit, (b) the maximum degree of super-saturation, and (c) the degree of under cooling at the throat. Assume $pv^{1.3}$ = constant for super-saturated flow.

[**Ans.** 8.4 cm^2, 1.65, 18°C]

6.10 Supersaturated expansion occurs in a nozzle supplied with steam at 20 bar and 325°C. Exit pressure is 3.6 bar. For

a flow rate of 7.5 kg/s, determine (a) the throat and exit areas and (b) the degree of undercooling at the exit.

[**Ans.** 28.9 cm^2, 42.8 cm^2, 10.9°C]

6.11 A converging nozzle has an exit area of 0.001 m^2. Air enters the nozzle with negligible velocity at a pressure of 1.0 MPa and a temperature of 360 K. For isentropic flow of an ideal gas with $k = 1.4$, determine the mass flow rate and the exit Mach number for back pressure of (a) 500 kPa and (b) 784 kPa.

[**Ans.** 2.13 kg/s, 1.79 kg/s]

6.12 Air flows through a convergent–divergent nozzle. At some section in the nozzle the pressure is 2 bar, the velocity is 171.1 m/s and the temperature is 1200 K. The area of flow at this section is

$1000\ cm^2$. Assuming isentropic flow conditions, determine the following:

1. Stagnation temperature and stagnation pressure.
2. Sonic velocity and Mach number at this section.
3. Velocity, Mach number, and area of flow at the exit section if the pressure at that section is 1.013 bar.
4. Pressure, temperature, velocity, and area of flow at the throat of the nozzle.

Assume c_p = 1.005 kJ/kg.K and c_v = 0.718 kJ/kg.K

[ESE, 1983]

6.13 A set of steam nozzles in an impulse turbine stage is supplied with steam at 20 bar and 230°C. The mass flow of steam is 60 kg/s. The steam is expanded from rest to a back pressure of 14 bar with an efficiency of 98%. The mean diameter of nozzle disc is 800 mm and the nozzle angle is 20 degrees. Assuming that steam is admitted all

around the periphery of the nozzle disc, determine:

1. whether a convergent or a convergent–divergent nozzle is to be used,
2. the velocity of flow of steam at the exit of the nozzle,
3. the area of flow normal to the axis of the stage, and
4. the height of the nozzles.

Assume adiabatic index for superheated steam to be 1.3. What would be the mass flow through the set of nozzles when the back pressure is reduced to 9 bar keeping the upstream conditions same as before?

[ESE, 1984]

6.14 Air is isentropically expanded in a convergent–divergent nozzle from an initial pressure of 5 bar and 25°C to a back pressure of 1.5 bar. The velocity of the air entering the nozzle is 100 m/s. The mass flow rate of the air is 2 kg/s.

Determine (a) Mach number at inlet to the nozzle, (b) pressure at the throat, (c) area of flow at the throat, and (d) the area of flow at the exit of the nozzle. Assume for air the ratio of specific heats to be 1.4 and R to be 0.287 kJ/kg.K.

[ESE, 1986]

6.15 Air enters a constant-area combustion chamber at a Mach number of 0.25 with a total pressure of 5.5 bar. Combustion gas leaves the chamber with a Mach number of 0.35. Neglecting friction and the mass of the fuel, determine the drop in total pressure of air in the combustion chamber. Assume $c_p/c_v = 1.35$.

[ESE, 1995]

ANSWERS TO MULTIPLE-CHOICE QUESTIONS

1. b
2. c
3. c
4. c
5. d
6. c
7. d
8. c
9. b
10. d
11. a
12. a
13. d
14. b
15. b
16. b
17. d
18. c
19. b
20. c
21. a
22. d
23. d
24. d
25. a
26. c
27. d
28. b
29. b
30. b
31. c
32. a
33. a
34. d
35. a
36. d
37. a
38. b
39. c
40. d
41. a
42. c
43. c

44. b
45. d
46. c
47. c
48. a
49. c
50. c
51. b
52. a
53. b
54. c
55. b
56. c
57. a
58. c
59. d
60. d
61. b

Chapter 7

Steam Turbines

7.1 ❒ PRINCIPLE OF OPERATION OF STEAM TURBINES

A steam turbine works on the dynamic action of steam. Steam is caused to fall in pressure in a passage or nozzle. Due to this fall in pressure, certain amount of heat energy is converted into kinetic energy to give it high velocity. The high velocity steam impinges on the moving blades of the turbine and changes the direction of motion and thus gives rise to a change in momentum, therefore, a force. Essentially, the nozzles direct the steam so that it flows in a well-formed high velocity jet. Moving buckets

convert this high-velocity jet to mechanical work in a rotating shaft.

In impulse turbines, there is a drop of steam pressure and consequent development of kinetic energy takes place solely in the stationary nozzles, and the work is obtained by the conversion of this kinetic energy into work on moving blades. In the reaction turbine, only a part of the kinetic energy conversion occurs in the stationary nozzles, whereas the remainder of the kinetic energy conversion is accomplished by a pressure drop in steam as it passes through the moving blades.

7.2 ❐ CLASSIFICATION OF STEAM TURBINES

On the basis of principles of operation,

steam turbines are classified as follows:

1. Impulse turbine
2. Reaction turbine
3. Impulse-reaction turbine

1. **Impulse Turbine:** This type of turbine is called a simple impulse turbine because the expansion of steam takes place only in one set of nozzles, as shown in Fig. 7.1. The pressure of steam falls from that in the steam nozzle to that existing in the condenser, whereas the steam flows through it. Hence, the pressure in the wheel chamber is practically equal to the condenser pressure.

   The high-velocity jet coming out from the nozzle impinges on the blades, so that the whole kinetic energy is converted into mechanical energy. In practice, this type of turbine is used for small power ratings. The rotor diameter is kept small and consequently, the rotational speed becomes very high. This is known as *De-Laval turbine.*

Figure 7.1 *Simple impulse turbine*

2. **Reaction Turbine:** In a reaction turbine, there are guide blades instead of nozzles which convert the pressure energy into kinetic energy. The steam passing over the moving blades has the difference of pressure at the inlet tip and exit tip; hence, there is a drop of pressure in steam while passing over the moving blades. The fixed blade serves the purpose of nozzles which changes the direction of steam and at the same time, allows it to expand to a higher velocity. The pressure of steam falls as it passes over the moving blades. The diameter of each stage of a reaction turbine must increase after each group of blade rings in order to accommodate the increased volume of steam at lower pressure.

Figure 7.2 *Impulse-reaction turbine*

3. **Impulse-Reaction Turbine:** If the pressure of steam at outlet from the moving blades of the turbine is less than that at the inlet side of the blades, the drop in pressure suffered by steam during its flow through the moving blades causes a further generation of kinetic energy within the blades and adds to the preparing force which is applied to the turbine shaft; such a turbine is called *impulse-reaction turbine*.

This type of turbine is shown in Fig. 7.2. In this type of turbine, there is application of both principles, namely, impulse and reaction. There are several rows of moving blades fixed to the shaft and equal number of fixed blades attached to the casing. The fixed blades correspond to the nozzles of an impulse turbine. Steam is admitted for the entire circumference and therefore, there is all-round admission. In passing through the first set of fixed blades, the steam undergoes a small pressure drop and the velocity is increased. It enters the first set of moving blades and suffers the change of direction and therefore momentum. This gives an impulse to the blades. However, here, the passage to the blades is so designed that there is also a small drop in pressure in the moving blades, giving rise to an increase in kinetic energy. This drop in pressure gives rise to reactions in the direction opposite to that of added velocity. Thus, the driving force is the vector sum of impulse and reaction turbines. Normally, the turbine is known as reaction turbine. This is also called *Parson's reaction turbine*.

7.3 ❐ COMPARISON OF IMPULSE AND REACTION TURBINES

The comparison of impulse and reaction turbines is given in Table 7.1.

Table 7.1 *Comparison of impulse and reaction turbines*

7.4 ❐ COMPOUNDING OF IMPULSE TURBINES

If the entire pressure drop from boiler pressure to condenser pressure is carried out in a single stage nozzle, the velocity of steam entering the turbine blades will be very high and consequently, the turbine speed will be very high. Such high speed turbine rotors are not useful for practical purposes and a reduction gearing is necessary between the turbine and the generator driven by the turbine. There is also a danger of structural

failure of the blades due to excessive centrifugal stresses. Further, the velocity of steam at the exit of the turbine is also high when a single stage of blades is used. This gives rise to considerable loss of kinetic energy, reducing the efficiency of the unit. This loss of kinetic energy is termed as ‘carry over losses’ or ‘leaving losses’.

Compounding is a method employed for reducing the rotational speed of the impulse turbine to practical limits by using more than one stage. There are three methods of compounding impulse turbines as follows:

1. Velocity compounding
2. Pressure compounding
3. Pressure and velocity compounding

1. **Velocity Compounding:** This system consists of a nozzle or a set of nozzles and a wheel fitted with two or more rows of

blades as shown in Fig. 7.3. It has two rings of moving blades and two rows of fixed or guide blades placed. Steam entering the nozzle expands from the initial pressure to the exhaust pressure. On passing through the first row of moving blades, the steam gives only a part of kinetic energy and issues from this row of blades with a fairly high velocity. It then enters the guide blade and the moving blade. There is a slight drop in velocity in the guide blade due to friction. While passing through the second row of moving blades, the steam suffers a change in momentum and gives a part of kinetic energy to rotor. The steam leaving from the second row of moving blades is redirected to the second row of guide blades. By doing work on the third row of moving blades, the steam finally leaves the wheel in more or less axial direction with a certain residual velocity, about 2% of the initial velocity of steam at nozzle exit. Thus, the whole of velocity is not utilised on one row of moving blades but in a number of stages, as in a *Curtis turbine*.

Figure 7.3 *Velocity-compounded impulse turbine*

2. **Pressure Compounding:** This arrangement consists of allowing the expansion of steam in a number of steps by having a number of simple impulse turbines in the series on the same shaft, as shown in Fig. 7.4. The exhaust steam from one turbine enters the nozzles of succeeding turbines. Each of the simple impulse turbines is called a 'stage' of the turbine comprising its sets of nozzles and blades. This is equivalent to splitting up the whole pressure drop into a series of smaller pressure drops, hence, the term 'pressure compounding'.

The pressure compounding causes a smaller transformation of heat energy into kinetic energy to take place in each stage than in a simple impulse turbine. Hence, the steam velocity is much lower, with the result that the blade velocity and rotational speed may be lowered. The leaving loss is only 1–2% of the total available energy. *Rateau turbine* is a pressure compounded turbine.

3. **Pressure and Velocity Compounding:** It is a combination of pressure compounding and velocity compounding. The total pressure drop of steam is divided into stages and the velocity of each stage is compounded. This allows a bigger pressure drop in each stage and hence, less stages are necessary which require a shorter turbine for a given pressure drop. Such a turbine is shown in Fig. 7.5. The diameter of such a turbine increases in each stage in order to accommodate for a larger volume of steam at the lower pressure. The pressure is constant in each stage.

Figure 7.4 *Pressure compounded impulse turbine*

Figure 7.5 *Pressure- and velocity-compounded impulse tur*bine

7.5 ❐ VELOCITY DIAGRAMS FOR IMPULSE STEAM TURBINE

Let u = circumferential or tangential linear velocity of blades

v_{a1} = absolute velocity of steam at inlet of moving blade

v_{a2} = absolute velocity of steam at outlet of moving blade

v_{w1} = velocity of whirl at the entry of moving blade

$= v_{a1} \cos \alpha_1$

v_{w2} = velocity of whirl at the exit of moving blade

$= v_{a2} \cos \alpha_2$

v_{f1} = velocity of flow at inlet of moving blade

$= v_{a1} \sin \alpha_1$

v_{f2} = velocity of flow at outlet of moving blade

$= v_{a2} \sin \alpha_2$

v_{r1} = relative velocity of steam at entrance to moving blade

v_{r2} = relative velocity of steam at exit

of moving blade

$\dot{m}$ = mass rate of flow of steam, kg/s.

D = diameter of blade drum.

h = height of blade.

α_1 = angle which the absolute velocity of steam at inlet makes with the plane of moving blades, or nozzle angle or outlet angle of fixed blades.

α_2 = angle which the absolute velocity at outlet makes with the plane of moving blades or inlet angle of fixed blade.

β_1 = inlet angle of moving blade.

β_2 = exit angle of moving blade.

The jet of steam impinges on the moving blades at angle α_1 to the tangent of wheel with a velocity v_{a1}. This velocity v_{a1} has the following components.

1. Tangential or whirl component v_{w1} and since it is in the same direction as the motion of blades, it is the actual component which does work on the blade.
2. Axial or flow component v_{f1} and since it is perpendicular to the direction of motion of blade, it does no work. However, this component is responsible for the flow of steam through the turbine. This component also causes axial thrust on the rotor.

The velocity diagrams at inlet and outlet of a moving blade are shown in Fig. 7.6(a). Figure 7.6(b) shows the combined velocity diagrams.

Figure 7.6 *Velocity diagrams for impulse turbine: (a) Velocity diagrams at inlet and exit, (b) Combined velocity diagrams*

For blades with a smooth surface, it can

be assumed that friction loss is very less or zero. However, there is always a certain loss of velocity during the flow of steam over the blade and this loss is taken into account by introducing a factor called *blade velocity coefficient*, *K*. It is given by

Blade velocity coefficient, $K = v_{r2}/v_{r1}$, where $v_{r2} < v_{r1}$.

Note that K accounts for the loss in relative velocity due to friction of blades.

1. **Power Developed by the Turbine:**

   Work done per kg of steam,

   w = Force in the direction of blade motion × Distance travelled in the direction of force

   = Rate of change of momentum × Distance travelled

$= [v_{w_1} - (-\, v_{w_2})] \times u$

$= (v_{w_1} + v_{w_2})\, u$

Power developed by the turbine for a mass rate of flow of $\dot{m}$ kg/s of steam,

2. **Diagram or Blade Efficiency, η_d or η_b:**

   For a single blade stage,

3. **Gross or Stage Efficiency, η_s:**

   Stage efficiency takes into account the losses in the nozzle.

4. **Axial Thrust:** The axial thrust on the wheel is generated due to the difference between the velocity of flow at inlet and outlet.

   Axial thrust, F_a = Mass rate of flow of steam × Change in axial velocity

5. **Energy converted into heat due to blade friction:**

   = Loss of kinetic energy during flow of steam over the blades

6.

7.5.1 Condition for Maximum Blade Efficiency

Blade efficiency,

From velocity diagrams given in Fig. 7.6(b), we have

where

Now $BE = AE - AB = v_{a1} \cos \alpha_1 - u$

Let speed ratio,

For η_b to be maximum,

$\therefore \cos \alpha_1 - 2\rho = 0$

For symmetrical blades, $\beta_1 = \beta_2$, so that $C = 1$ and with no friction over the blades, $K = 1$

7.5.2 Maximum Work Done

Now, $w = (v_{w1} + v_{w2})u$

Equations (7.11) and (7.12) represent parabola. The blade efficiency has been plotted against speed ratio in Fig. 7.7, without losses and with losses for $\alpha_1 = 20°$, $\beta_1 = \beta_2$ and $K = 0.85$.

Figure 7.7 *Blade efficiency v's speed ratio*

7.5.3 Velocity Diagrams for Velocity Compounded Impulse Turbine

The velocity diagrams for the first and second stage moving blades of a velocity-compounded impulse turbine are shown in Fig. 7.8(a) and Fig 7.8(b), respectively. Consider that the final absolute velocity of steam leaving the second row is axial. The velocity u of the blades for both the rows is the same

as they are mounted on the same shaft and are of equal height.

Work done in the first row of moving blades,

$$w_1 = u(v_{w1} + v_{w2}) = u(v_{r1} \cos \beta_1 + v_{r2} \cos \beta_2)$$

If there is no friction loss, $v_{r1} = v_{r2}$ and for symmetrical blades, $\beta_1 = \beta_2$.

$$\therefore w_1 = 2u\, v_{r1} \cos \beta_1 = 2u(v_{r1} \cos \alpha_1 - u)$$

Now, $v_{a3} = v_{a2}$

Work done in the second row of moving blades,

$$w_2 = u\, v_{w3} \text{ as } v_{w2} = 0 \text{ and } a_4 = 90°.$$
$$= u\,(v_{r3} \cos \beta_3 + v_{r4} \cos \beta_4)$$

Figure 7.8 *Velocity diagrams for velocity compounded impulse turbine: (a) First stage, (b) Second stage*

For no friction, $v_{r3} = v_{r4}$, and for symmetrical blades, $\beta_3 = \beta_4$.

$\therefore w_2 = 2uv_{r3} \cos \beta_3 = 2u\,(\mathrm{v}_{a3} \cos \alpha_3 - u)$

For $\alpha_3 = \alpha_2$

$$v_{a3} \cos \alpha_3 = v_{a2} \cos \alpha_2$$
$$= v_{r2} \cos \beta_2 - u = v_{r1} \cos \beta_1 - u$$
$$= (v_{a1} \cos \alpha_1 - u) - u = v_{a1} \cos \alpha_1 - 2u$$

$\therefore w_2 = 2u\,[(v_{a1} \cos \alpha_1 - 2u) - u] = 2u\,(v_{a1} \cos \alpha_1 - 3u)$

Total work done, $w_t = \mathrm{w}_1 + w_2$

$$= 2u\,(v_{a1}\cos\alpha_1 - u) + 2u\,(v_{a1}\cos\alpha_1 - 3u)$$
$$= 2u\,(2v_{a1}\cos\alpha_1 - 4u)$$
$$= 4u\,(v_{a1}\cos\alpha_1 - 2u)$$

Blade efficiency, $\eta_b =$

For η_b to be maximum,

$$\cos\alpha_1 - 4\rho = 0$$

For

where n = number of rotating blade rows in series.

7.5.4 Effect of Blade Friction on Velocity Diagrams

In an impulse turbine, the relative velocity at the outlet will be the same as

the relative velocity at the inlet, if friction is neglected. In practice, there is a frictional resistance to the flow of steam jet over the blade, the effect of which is to cause a slowing down of the relative velocity. Usually, there is a loss of 10–15% in the relative velocity due to friction. Owing to frictional resistance of the blades, it will be found that $v_{r2} = Kv_{r1}$, where K is a coefficient, which takes the blade loss due to friction into account. The velocity diagrams considering blade friction are shown in Fig. 7.9. Here, the inlet diagram is first drawn and the line BC, of an unknown length, is drawn at the correct angle β_2. Mark off on line $BD = v_{r1}$, the friction loss of relative velocity DD', then $BD' = Kv_{r1}$. With B as centre, draw an arc of

radius BD' to cut BC at C. Then $BC = v_{r2} = Kv_{r1}$.

By joining A and C, the line AC representing v_{a2} is obtained. This completes the outlet velocity diagram.

Figure 7.9 *Velocity diagrams for impulse turbine considering blade friction*

7.5.5 Impulse Turbine with Several Blade Rings

The stage of an impulse turbine, which is compounded for velocity, consists of alternate rings of nozzles, and moving and fixed blades. As the blade velocity u is constant for all the moving blade rings of the stage, the velocity diagrams for all the blade rings can be superimposed on the same base, represented by u. Figure 7.10 shows the velocity diagram for a stage consisting

of two moving and one fixed blade rings. Let us assume that the following data is known:

1. Blade velocity, u.
2. Nozzle angle, α_1.
3. The moving blade angles, β_1 and β_2, which are assumed to be same for both blade rings.
4. The velocity of steam v_{a1} discharged from nozzle.
5. Blade friction loss, 10%, i.e., $K = 0.9$.

The following steps may be followed for drawing the velocity diagrams:

1. Draw $AB = u$ to any convenient scale. Draw AD inclined at angle α_1 to AB so that $AD = v_{a1}$ to the same scale. Join BD. Then BD = v_{r1} for the first moving blade ring.
2. Mark off $DD' = 0.1 \times BD$ so that $v_{r2} = 0.9\ v_{r1} = BD'$. With B as centre, draw an arc of radius BD' to cut BC at C where the line BC is drawn at angle of β_2 to BA. Join AC. Then $AC = v_{a2}$ for the first moving blade ring.
3. The steam now flows over the fixed blade ring and will lose 10% of its velocity during the passage. Hence, mark off $CC' = 0.1 \times AC$. With A as centre, draw on arc of radius AC' to cut BD at G. Then $AG = v_{a3}$ representing the steam velocity when entering the second moving blade ring. The velocity diagram for the second blade ring is triangle AGB. BG = v_{r3}, the relative velocity.

Figure 7.10 *Velocity diagrams for impulse turbine considering friction with several blade rings*

4. The steam now flows over the second moving blade and loses one-tenth of its relative velocity due to friction., Hence, mark off $GG' = 0.1 \times BG$ so that $BG' = 0.9\ BG$. With B as centre and radius BG', draw an arc to cut BC at H, then $AH = v_{a4}$ and BH

$= v_{r4}$.

α_2 = angle of discharge from first moving blade

α_4 = angle of discharge from second moving blade

α_3 = outlet angle of fixed blade.

Work done per kg of steam for first moving blade ring,

$$w_1 = (v_{w_1} + v_{w_2})u$$

Work done per kg of steam for second moving blade ring.

$$w_2 = (v_{w_3} + v_{w_4})u$$

Total Work done, $w_t = w_1 + w_2$

$$= [(v_{w_1} + v_{w_2}) + (v_{w_3} + v_{w_4})]u$$

Power developed per stage =

Blade efficiency, η_b =

Stage efficiency, η_s =

Total axial thrust =

7.6 ❐ ADVANTAGES AND LIMITATIONS OF VELOCITY COMPOUNDING

7.6.1 Advantages

1. Fewer number of stages are required and therefore, the initial cost is less.
2. The space required is less.
3. The pressure in the housing is considerably less which requires cheaper and thin casing for the turbine.

7.6.2 Limitations

1. The friction losses are larger due to the high velocity of steam.
2. The maximum blade efficiency and efficiency range decrease with the increase in the number of stages.
3. The power developed in each successive blade row decreases with increase in number of rows.
4. All the stages are not used with equal economy.

7.7 ❐ VELOCITY DIAGRAMS FOR IMPULSE-REACTION TURBINE

The steam passes to the fixed blades with velocity v_{a1}. Fixed blades also act as nozzles with pressure drop occurring, while steam passes through them so that there is gain in kinetic energy and steam leaves the fixed blades with velocity v_{a2}.

The expansion of steam and the enthalpy drop occur in fixed and moving blades. The velocity diagrams for an impulse-reaction turbine are shown in Fig. 7.11.

1. **Degree of Reaction:** The degree of reaction R_d of a reaction turbine is defined as the ratio of enthalpy drop over moving blades to the total enthalpy drop in the stage. Thus

   Enthalpy drop through fixed blades,

   Enthalpy drop along the moving blades, $\Delta h_m =$

   Kinetic energy supplied to moving blades, K.E.

   Neglecting friction of blade surface,

   Axial thrust on rotor = $(v_{f1} - v_{f2}) + (\Delta p)_m \times$ area of blade disc

   Now total enthalpy drop in the stage = Work done by steam in the stage

$$\Delta h_f + \Delta h_m = u\,(v_{w1} + v_{w2})$$

Figure 7.11 *Velocity diagrams for impulse-reaction turbine*

From Fig. 7.11, we have

$$v_{r2} = v_{f2} \text{ cosec } \beta_2 \text{ and } v_{r1} = v_{f1} \text{ cosec } \beta_1$$

and $v_{w1} + v_{w2} = v_{f1} \cot \beta_1 + v_{f2} \cot \beta_2$

The velocity of flow generally remains constant through the blades.

$\therefore v_{f1} = v_{f2} = v_f$

If $R_d = 0.5$, then

From Fig. 7.11, we have

$$u = v_f(\cot \beta_2 - \cot \alpha_2) = v_f(\cot \alpha_1 - \cot \beta_1)$$

This means that the moving and fixed blades must have the same shape if the degree of reaction is 50%. This condition gives symmetrical velocity diagrams. This type of turbine is known as *Parson's Reaction Turbine*.

2. **Efficiency:** We assume that the degree of reaction is 50%, that is, $\Delta h_f = \Delta h_m$, the moving and fixed blades are of the same shape, and the velocity of steam at exit from the preceding stage is the same as the velocity of steam at the entrance to the succeeding stage.

The work done per kg of steam, $w = u(v_{w1} + v_{w2})$
$= u\,[\mathrm{v}_{a1} \cos \alpha_1 + (v_{r2} \cos \beta_2 - u)]$

Now, $\beta_2 = \alpha_1$ and $v_{r2} = v_{a1}$

where is the speed ratio.

K.E. supplied to fixed blade =

K.E. supplied to moving blade

Total energy supplied to stage,

For symmetrical blades, $v_{r2} = v_{a1}$

From Fig. 7.11, we have

For η_b to be maximum,

$$(1 + 2\rho \cos \alpha_1 - \rho^2)(4 \cos \alpha_1 - 4\rho) - 2\rho(2 \cos \alpha_1 - \rho)(2 \cos \alpha_1 - \rho) = 0$$

$$4(\cos \alpha_1 - \rho)(1 + 2\rho \cos \alpha_1 - \rho^2) - 4\rho(\cos \alpha_1 - \rho)(2 \cos \alpha_1 - \rho) = 0$$

$$(\cos \alpha_1 - \rho)[(1 + 2\rho \cos \alpha_1 - \rho^2) - \rho(2 \cos \alpha_1 - \rho)] = 0$$

$\therefore \cos \alpha_1 - \rho^2 = 0$

3. **Height of Turbine Blading:**

The height and thickness of blading are shown in Fig. 7.12.

Volume flow = Area × flow velocity

where d = mean diameter of turbine wheel

h_1, h_2 = blade heights at inlet and outlet respectively

t_1, t_2 = thickness of blades at inlet and outlet respectively.

n = number of blades

Figure 7.12 *Height of turbine blading*

For most of the turbines,

$$v_{f1} = v_{f2} = v_f,\ h_1 = h_2 = h \text{ and } t_1 = t_2 = t$$

Equation (7.31) gives height of blades.

The pitch of blades, p is given by:

$$\dot{m}v_s = [n\,(p - t)h]\ v_f$$

Generally $t << p$,

$$\therefore \dot{m}v_s = n\,ph\,v_f = \pi\,dhv_f$$

7.8 ❒ REHEAT FACTOR

The expansion of steam through a number of stages of turbine is shown in Fig. 7.13. A_1B_1 represents isentropic expansion in the first stage. The actual state of steam with frictional reheating is shown by point A_2. Therefore, the actual heat drop in the first stage is A_1C_1. Similarly, the isentropic and actual heat drops in the succeeding stages are shown by A_2B_2, A_3B_3, ….., A_2C_2, A_3C_3, ….., and so on. A_1D represents the isentropic heat drop between supply and condenser state of steam.

Stage efficiency,

For first stage,

Figure 7.13 *Expansion of steam through a number of turbine stages*

For second stage, and so on.

Turbine internal efficiency

If $\eta_{s1} = \eta_{s2} = \ldots.. = \eta_{s'}$ then

From Eqs. (7.34) and (7.35), we get

Combining Eqs (7.33) and (7.36), we get

$$\eta_i = \eta_s R_f$$

Reheat factor is always greater than one. Normally R_f lies between 1.02 to 1.05. The line passing through A_1, A_2, … is called the *condition curve*.

7.9 ❐ LOSSES IN STEAM TURBINES

The various losses in steam turbines and their causes are as follows:

1. **Residual Velocity Loss:** This loss occurs due to the absolute exit velocity of steam and is equivalent to where v_{a2} is the absolute exit velocity of steam. In a single stage turbine, it may be about 10–12% and can be reduced by using multi-stages.
2. **Frictional and Turbulence Loss:** Friction loss mainly occurs in nozzles and turbine blades. The nozzle efficiency is used to account the friction loss. The loss due to friction and turbulence is about 10%.
3. **Leakage Loss:** It occurs at the following points:
    1. Between the turbine shaft and bearings
    2. Between the shaft and stationary diaphragms carrying nozzles, and blade tips
    3. Through the labyrinth glands

   The total leakage loss is about 1–2%.

4. **Mechanical Friction Loss:** Friction between shaft, bearing, and regulating valves accounts for this loss. It can be reduced by proper lubrication.
5. **Wet Steam Loss:** The velocity of water particles is less than that of steam; therefore, water particles have to be dragged with the steam causing loss of kinetic energy.
6. **Radiation Loss:** This is due to much higher temperature of turbine as compared to the surroundings. This loss can be reduced by proper insulation.

7.10 ❐ TURBINE EFFICIENCIES

1. Blade or diagram efficiency, η_b
2. Stage efficiency,

3. Internal efficiency,
4. Overall or turbine efficiency,

5. Net efficiency or efficiency ratio,

7.11 ❐ GOVERNING OF STEAM TURBINES

The purpose of governing of steam turbines is to maintain the speed of a turbine sensibly constant, irrespective of the load. The various methods of governing are as follows:

1. Throttle governing
2. Nozzle control governing
3. By-pass governing
4. Combination of throttle governing and nozzle control governing
5. Combination of throttle governing and by-pass governing

1. **Throttle governing:** The line diagram of throttle governing is shown in Fig. 7.14. The steam flow to the turbine is throttled by a balanced throttle valve actuated by a centrifugal governor. An oil differential relay is incorporated to magnify the small force produced by the governor for a small change of speed to actuate the throttle valve. The throttle valve is moved by a relay piston. A floating differential lever is fixed to its one end and a piston valve is fixed at some intermediate point. The pilot piston valve consists of two piston valves covering ports without any overlap. The piston valves are also operated by lubricating oil supplied by a pump at about 3 to 4 bar. The oil from this chamber is returned by the oil drain.

   Operation: Let the turbine work at full-rated load at the rated constant speed. If the load is reduced, the energy supplied to the turbine will

be in excess and the turbine rotor will accelerate. Thus, the governor sleeve will lift. As the throttle valve position is assumed to be the same momentarily, the pilot piston valve spindle will get lifted, opening the upper port to oil pressure and lower port to oil return. The relay piston will thus close the throttle valve partially. The lowering of the throttle valve spindle will lower the pilot piston spindle and close the ports. As soon as the ports are closed, the relay piston gets stabilised in one position corresponding to the reduced load. The change in the available enthalpy for throttle governing is shown in Fig. 7.15.

2. **Nozzle control governing:** The principle of nozzle control governing is accomplished by uncovering as many steam passages as are necessary to meet the load by poppet valves. An arrangement is shown in Fig. 7.16. The nozzles are divided into groups N_1, N_2, and N_3 under the control valves V_1, V_2, and V_3, respectively. The number of nozzle groups may vary from three to five or more.
3. **By-pass governing:** The principle of by-pass governing is shown in Fig. 7.17. Steam entering the turbine passes through the main throttle valve which is under the control of the speed governor and enters the nozzle box or the steam chest. For loads greater than the economical load, a by-pass valve is opened, allowing steam to pass from the first stage, nozzle box into the steam chest, and so into the nozzles of the fourth stage. The by-pass valve is not opened until the lift of the throttle valve exceeds a certain amount. As the load diminishes, the by-pass valve closes first. The by-pass valve is under the control of speed governor for all loads.

Figure 7.14 *Throttle governing*

Figure 7.15 *T-s diagram for throttling*

Figure 7.16 *Nozzle governing*

Figure 7.17 *By-pass governing*

7.12 ❐ LABYRINTH PACKING

Labyrinth seals are characterised as controlled clearance seals without rubbing contact with the moving parts and with some tolerable leakage. The fluid throttling is achieved in steps, using a series of small chambers, where a sudden irreversible acceleration with subsequent deceleration of the leaking fluid takes place. Every step down in this process of pressure dissipation is accompanied by a loss in pressure. Due to the lack of direct rubbing contact with the moving shaft, the labyrinth seal is well suited for sealing shafts operating at high rotational speeds, as in centrifugal compressors and steam

turbines. It requires neither lubrication nor maintenance. The simplest design of a labyrinth packing is with a straight shaft and a straight housing is shown in Fig. 7.18(a). The modified labyrinth designs are of the staggered, stepped, and interference configurations (Figs. 7.18(b)–(e)).

Figure 7.18 *Labyrinth seal designs: (a) Straight shaft and straight housing, (b) Staggered, (c) Staggered and stepped, (d) Stepped, (e) Interference*

7.13 ❒ BACK PRESSURE TURBINE

There are several industries, such as paper making, textile, chemical, dyeing, sugar refining, and so on, in which there is a dual demand for power and steam for heating and process work. Producing steam separately for power and heating is wasteful. If the turbine is operated with normal exhaust pressure and the

temperature of the exhaust steam is too low to be of any use for heating purposes, by suitable modification of the initial and exhaust pressures, it would be possible to generate the required power and still have available for process work a large quantity of heat in the exhaust steam.

Figure 7.19 *Back pressure turbine plant*

Figure 7.20 *Thermodynamics of back pressure turbine*

The back pressure turbine may be used in cases where the power, which may be generated by expanding steam from an economical initial pressure down to the heating pressure, is equal to, or greater than the power requirements. The layout of such a plant is shown in Fig. 7.19.

Steam is generated in the boiler at a

suitable working pressure and admitted to the turbine. The exhaust steam from the turbine will normally be superheated and in most cases, is not suitable for process work. A de-superheater is used to make it suitable for process work by spraying a jet of water, thermostatically controlled, on the entering steam. The steam is cooled and the spray water evaporated. The new saturated steam enters the heaters and is entirely condensed. The condensed steam may or may not be returned to the boiler. The thermodynamics of back pressure turbine is shown in Fig. 7.20.

7.14 ❐ PASS OUT OR EXTRACTION TURBINE

In many cases, the power available from a back pressure turbine through which

the whole of the heating steam flows is appreciably less than that required in the factory. In such cases, it would be possible to install a back pressure turbine to generate additional power. However, generally, the functions of both machines are combined in a single turbine. Such a turbine is shown in Fig. 7.21. The steam enters the turbine and expands through the high pressure stages before the extraction branch. Here, a certain quantity of steam is continuously being extracted for heating purposes, the remainder passes through the pressure control valve into the low-pressure part of the turbine.

Figure 7.21 *Pass out or extraction turbine*

7.15 CO-GENERATION

The exhaust steam from a steam power plant is often rejected to the atmosphere. When the heat of exhaust steam is used for heating of buildings, and heating required by many industrial processes, the functions of heating and power production can often be combined effectively. This combination is often called *co-generation* (Fig. 7.22). The process heater replaces the condenser of an ordinary Rankine cycle. The pressure at the exhaust from the turbine is the saturation pressure corresponding to the temperature desired in the process heater. Such a turbine is called *back pressure turbine*.

Figure 7.22 *Co-generation*

Example 7.1

Steam at 4.9 bar and 160°C is supplied to a single stage impulse turbine at the rate of 60 kg/min. From there, it is exhausted to a condenser at a pressure of 0.196 bar. The blade speed is 300 m/s. The nozzles are inclined at 25° to the plane of the wheel and outlet blade angle is 35°. Neglect friction losses and estimate (*a*) the theoretical power developed by the turbine, (*b*) the diagram efficiency, and (*c*) the stage efficiency. **[IES, 1990]**

Solution

Given that $p_1 = 4.9$ bar, $t_1 = 160$°C, $\dot{m} = 60$ kg/min, $p_2 = 0.196$ bar, $u =$ 300 m/s, $a_1 = 25°$, $\beta_2 = 35°$, $K = 1$

From Mollier diagram, at 4.9 bar and 160°C, $h_1 = 2700$ kJ/kg

Assuming isentropic expansion, at 0.196 bar, $h_2 = 2200$ kJ/kg

Inlet velocity of steam to turbine = outlet velocity of steam from nozzle

The velocity diagrams are shown in Fig. 7.23.

$$v_{w1} = v_{a1} \cos \alpha 1 = 1000 \cos 25° = 906.3 \text{ m/s}$$
$$v_{w2} = v_{r2} \cos \beta_2 - u$$
$$v_{r2} = v_{r1} \text{ for } K = 1$$

or $v_{r1} = 739$ m/s

$$v_{w2} = 739 \cos 35° - 300 = 305.4 \text{ m/s}$$

1. Theoretical power developed

Figure 7.23 *Velocity diagrams for a single stage impulse turbine*

2. Blade efficiency, $\eta_b = 0.727$ or 72.7%
3. Stage efficiency, η_s = Work output/Heat input
$= 363.5/(2700 - 2200) = 0.727$ or 72.7%

Example 7.2

The first stage of an impulse turbine is compounded for velocity and has two rings of moving blades and one ring of fixed blades. The nozzle angle is 20° and the leaving angles of the blades are respectively: first moving 20°, fixed 25° and second moving 30°. The velocity of steam leaving the nozzle is 600 m/s, the blade speed is 125 m/s, and the steam velocity relative to the blades is reduced by 10% during the passage through each ring. Find the diagram efficiency under these conditions and the power developed

for a steam of 4 kg/s.

Solution

Given that u = 125 m/s, v_{a1} = 600 m/s, K = 0.9, α_1 = 20°, β_2 = 20°, β_3 = 25°, β_4 = 30°, $\dot{m}$ = 4 kg/s

Choose a scale of 1 cm = 50 m/s. The velocity diagrams are drawn in Fig. 7.24.

$$ab = u = 125 \text{ m/s} = 2.5 \text{ cm}$$
$$\angle bad = 20°,\ ad = v_{a1} = 600 \text{ m/s} = 12 \text{ cm}$$

Join bd. By measurement, bd = 9.7 cm, bd' = 9.7 × 0.9 = 8.7 cm

$$\angle abc = 20°,\ bc = bd'$$

Join ac. By measurement, ac = 6.4 cm, ac' = 0.9 × 6.4 = 5.7 cm

Figure 7.24 *Velocity diagrams for a two stage impulse turbine*

$$\angle bag = 25°$$
$$bg = ac'$$
$$bg = 3.6 \text{ cm}, \; bg' = 0.9 \times 3.6 = 3.2 \text{ cm}$$
$$\angle abh = 30°, \; bh = bg'$$

Join *ah*

$$v_{w1} = af = 11.3 \text{ cm}, \; v_{w2} = ea = 5.8 \text{ cm}, \; v_{w3} = ak = 5.1 \text{ cm},$$
$$v_{w4} = aj = 0.4 \text{ cm}$$

Blade efficiency, η_b =

Power developed =

Example 7.3

In an impulse turbine, the mean diameter of the blades is 1.05 m and the speed is 3000 rpm. The nozzle angle is 18°, speed ratio is 0.42, and

the friction factor is 0.84. The outlet blade angle is to be made 3° less than the inlet angle. The steam flow is 10 kg/s. Draw the velocity diagrams for the blades and calculate (a) the tangential thrust, (b) the axial thrust, (c) the resultant thrust, (d) the power developed, and (e) the blading efficiency.

Solution

Given that $d_m = 1.05$ m, $N = 3000$ rpm, $\alpha_1 = 18°$, $\rho = 0.42$, $K = 0.84$, $\beta_2 = \beta_1 - 3°$, $\dot{m} = 10$ kg/s

Blade speed, $u =$

Draw the velocity diagrams as shown in Fig. 7.25.

Figure 7.25 *Velocity diagrams for an impulse turbine*

or $\beta_1 = 30.2°$

$$\beta_2 = 30.2 - 3 = 27.2°$$
$$v_{r2} = K\, v_{r1} = 0.84 \times 241.28 = 202.67 \text{ m/s}$$
$$v_{f2} = v_{r2} \sin \beta_2 = 202.67 \sin 27.2° = 92.64 \text{ m/s}$$
$$v_{w1} = v_{a1} \cos \alpha_1 = 392.7 \cos 18° = 373.48 \text{ m/s}$$
$$v_{w2} = v_{r2} \cos \beta_2 - u = 202.67 \cos 27.2° - 164.93 = 15.33 \text{ m/s}$$

1. Tangential thrust $\dot{m} = (v_{w1} + v_{w2}) = 10\,(373.48 + 15.33) = 3888$ N
2. Axial thrust = $\dot{m}\,(v_{f1} - v_{f2}) = 10\,(121.35 - 92.64) = 287.1$ N
3. Resultant thrust = $[(3888)^2 + (287.1)^2]^{1/2} = 3898.6$ N
4. Power developed
5. Blade efficiency, η_b =

Example 7.4

One stage of impulse turbine consists of a converging nozzle ring and moving blades. The nozzles are

inclined at 22° to the blades whose tip angles are both 35°. If the velocity of steam at exit from nozzle is 660 m/s, find the blade speed so that the steam passes on without shock. Find the diagram efficiency neglecting losses if the blades are run at this speed.

[IES, 1992]

Solution

Given that $\alpha_2 = 22°$, $\beta_1 = \beta_2 = 35°$, $v_{a1} = 660$ m/s

The velocity diagrams are shown in Fig. 7.26.

$$v_{f1} = v_{a1} \sin 22° = v_{r1} \sin 35°$$

Figure 7.26 *Velocity diagrams for a single stage impulse turbine*

$$v_{w1} = v_{a1} \cos \alpha_1 = 660 \cos 22° = 611.94 \text{ m/s}$$

or $(431.05)^2 = u^2 + (660)^2 - 2u \times 660 \cos 22°$

or $185804 = u^2 + 435600 - 1223.88\, u$

or $u^2 - 1223.88\, u + 249796 = 0$

For $K = 1$, $v_{r2} = v_{r1} = 431.05$ m/s

$$v_{w2} = v_{r2} \cos \beta_2 - u = 431.05 \cos 35° - 258.85 = 94.24 \text{ m/s}$$

Example 7.5

A single-stage steam turbine is supplied with steam at 5 bar, 200°C

at the rate of 50 kg/min. It expands into a condenser at a pressure of 0.2 bar. The blade speed is 400 m/s. The nozzles are inclined at an angle of 20° to the plane of the wheel and the outlet blade angle is 30°. Neglecting friction losses, determine the power developed, blade efficiency, and stage efficiency.

[IES, 1994]

Solution

Given that p_1 = 5 bar, t_1 = 200°C, p_2 = 0.2 bar, $\dot{m}$ = 50 kg/min, u = 400 m/s, α_1 = 20°, β_2 = 30°, $v_{r1} = v_{r2}$ as $K = 1$

From steam tables,

At 5 bar, 200°C, $h_1 = 2855.4$ kJ/kg, $s_1 = 7.0592$ kJ/kg.K

At 0.2 bar, $s_{f2} = 0.8319$ kJ/kg. K, $s_{fg2} = 7.0766$ kJ/kg.K

$$s_2 = s_1 = s_{f2} + x_2 s_{fg2}$$

or $7.0592 = 0.8319 + 7.0766\, x_2$

or $x_2 = 0.88$

$$h_2 = h_{f2} + x_2\, h_{fg2} = 251.38 + 0.88 \times 2358.3 = 2326.68 \text{ kJ/kg}$$

Enthalpy drop, $\Delta h = h_1 - h_2 = 2855.4 - 2326.68 = 528.7$ kJ/kg

or $\beta_1 = 31.84°$

$$v_{r2} = v_{r1} = 655.6 \text{ m/s}$$
$$v_{w2} = v_{r2} \cos \beta_2 - u = 665.6 \cos 30° - 400 = 177 \text{ m/s}$$
$$v_w = v_{w1} + v_{w2} = 965.3 + 177.3 = 1143.6 \text{ m/s}$$

Power developed,

Blade efficiency, $\eta_b = = 0.8652$ or 86.52%

Example 7.6

A simple impulse turbine has a mean blade speed of 200 m/s. The nozzles are inclined at 20° to the plane of rotation of the blades. The steam velocity from nozzle is 600 m/s. The turbine uses 3600 kg/h of steam. The absolute velocity at exit is along the axis of the turbine. Determine: (a) the inlet and exit angles of blades, (b) the power output of the turbine, (c) the diagram efficiency, and (d) the axial thrust (per kg steam per second). Assume inlet and outlet angles to be

equal.

[IES, 1998]

Solution

Given that $u = 200$ m/s, $\alpha_1 = 20°$, $v_{a1} = 600$ m/s, $\dot{m} = 3600$ kg/h, $v_{a2} = v_{f2}$

The velocity diagrams are shown in Fig 7.27.

1. $v_{f1} = v_{a1} \sin \alpha_1 = 600 \sin 20° = 205.2$ m/s

$v_{w1} = v_{a1} \cos \alpha_1 = 600 \cos 20° = 563.8$ m/s

Figure 7.27 *Velocity diagrams for a simple impulse turbine*

or $b_1 = 29.42°$

$$\beta_2 = \beta_1 = 29.42°$$
$$v_{w2} = 0$$

2. Power output of turbine =
3. Diagram efficiency, = = 0.6264 or 62.64%
4. Axial thrust, $F_a = \dot{m}(v_{f1} - v_{f2})$

$$v_{f2} = u \tan \beta_2 = 200 \tan 29.42° = 112.79 \text{ m/s}$$
$$F_a = 1 \times (205.2 - 112.79) = 92.41 \text{ N}$$

Example 7.7

Steam at 2.94 bar absolute and dry saturated comes out of rotor of a reaction turbine having identical bladings. The velocity of steam entering into the turbine is 100 m/s. The mean blade height is 4 cm and the exit angle of the moving blade is 20°. The blade velocity is 4/3 times of axial flow velocity at the mean radius. If the steam flow rate is 2.5 kg/s, find (a) the rotor speed, (b) the power developed, (c) the diagram efficiency, (d) the percentage increase in relative velocity in the moving blades and (e) the enthalpy drop of steam in blade passage.

[IES, 1999]

Solution

The velocity triangles are shown in Fig. 7.28.

Given that $p = 2.94$ bar, $h = 4$ cm, $v_{a1} = 100$ m/s, $\dot{m} = 2.5$ kg/s, $\alpha_1 = \beta_2 = 20°$, $\beta_1 = \alpha_2$, $u = (4/3)v_f$

From velocity triangles, we have (Fig. 7.28)

$$v_f = v_{f1} = v_{f2} = v_{a1} \sin \alpha_1 = 100 \sin 20° = 34.2 \text{ m/s}$$

Specific volume of steam at 2.94 bar (from steam tables), $v_s = 0.6173$ m^3/kg

Figure 7.28 *Velocity diagrams for a reaction turbine having symmetrical blades*

$$v_s = \pi d h v_f$$

1.

2. Power developed, $P = \dot{m}u\,(v_{w1} - v_{w2})$

3. Diagram efficiency
4.

Increase in relative velocity =

5. Enthalpy drop:

Fixed blades,

Moving blades,

Example 7.8

In a multi-stage Parson's reaction turbine, at one of the stages, the rotor diameter is 125 cm and the speed ratio 0.72. The speed of the rotor is 3000 rpm. Determine: (a) the blade inlet angle if the blade outlet angle is 22°, (b) diagram efficiency, and (c) percentage increase in diagram efficiency and rotor speed if the turbine is designed to run at the best theoretical speed.

[IES, 2000]

Solution

Given that $d = 1.25$m, $N = 3000$ rpm, = 0.72, $\alpha_1 = \beta_2 = 22°$

1. m/s

   In Parson's reaction turbine, (see Fig. 7.29)

$$v_{r2} = v_{a1},\ v_{r1} = v_{a2} \text{ and } a_2 = b_1$$
$$v_{f1} = v_{a1} \sin \alpha_1 = 272.7 \sin 22° = 102.15 \text{ m/s}$$
$$v_{w1} = v_{a1} \cos \alpha_1 = 272.7 \cos 22° = 252.84 \text{ m/s}$$

   or $\beta_1 = 61°$

$$v_{r2} = 272.7 \text{ m/s}$$

Figure 7.29 *Velocity diagrams for Parson's reaction turbine*

2. $v_{w2} = v_{r2} \cos \beta_2 - u = 272.7 \cos 22° - 196.35 = 56.5$ m/s

   Diagram efficiency,

3. Power developed,kW

   At best theoretical rotor speed, $u = v_{a1} \cos \alpha_1 = 272.6 \cos 22° = 252.84$ m/s

Percentage increase in rotor speed = = 0.2877
or 28.77%

Diagram efficiency =

Percentage increase in diagram efficiency =

Example 7.9

A De-Laval turbine has a mean blade velocity of 180 m/s. The nozzles are inclined at 17° to the tangent. The steam flow velocity through the nozzles is 550 m/s. For a mass flow of 3300 kg/hr and for axial exit conditions, find (a) the inlet and outlet angles of the blade system, (b) the power output, and (c) the diagram efficiency.

[IES, 2001]

Solution

Given that $u = 180$ m/s, $v_{a1} = 550$m/s, $\dot{m} = 3300$ kg/h, $\alpha_1 = 17°$, $\alpha_2 = 90°$

The velocity diagrams are shown in Fig. 7.30.

1. $v_{f1} = v_{a2}$ (see Fig. 7.30)

   $v_{f1} = v_{a1} \sin \alpha_1 = 550 \sin 17° = 160.8$ m/s

   $\tan \beta_2 = v_{f1}/u = 160.8/180 = 0.89336$

   or $\beta_2 = 41.78°$

$$\tan \beta_1 = v_{f1}/(v_{a1} \cos \alpha_1 - u)$$
$$= 160.8/(550 \cos 17° - 180) = 0.46478$$

   or $\beta_1 = 24.93°$

Figure 7.30 *Velocity of diagrams for De-Laval turbine*

2. Power output, $P = \dot{m} u v_{w1} = (3300/3600) \times (180 \times 550 \cos 17°/10^3) = 85.785$ kW
3. Diagram efficiency = = $2 \times 180 \times 550 \cos 17°/550^2 = 0.626$ or 62.6%

Example 7.10

A steam turbine is governed by throttling. The full load output of a steam turbine, measured at the coupling is 5 MW and the losses due to bearing friction, the governor and oil pump drive, etc., are 200 kW. The steam is supplied at a pressure of 20 bar and 300°C. The exhaust pressure of steam is 0.07 bar. The internal efficiency ratio at full load is 0.75. Calculate the coupling power of turbine when the steam flow through the turbine is 20% of that at full load. Assume that the external losses are the same as that at full load, exhaust pressure is the same, and the internal efficiency ratio is reduced to 70% at

this load.

[IES, 2005]

Solution

The h–s (Mollier) diagram is shown in Fig 7.31.

At 20 bar, 300°C, $h_1 = 3020$ kJ/kg

At 0.07 bar, $h_2 = 2100$ kJ/kg

At full load:

Internal efficiency,

or

or $h_{2'} = 2330$ kJ/kg

$$\Delta h_i = 3020 - 2330 = 690 \text{ kJ/kg}$$

Internal power developed,

or $5000 = \dot{m} \times 690$

or $\dot{m} = 7.246$ kg/s

Nozzle box pressure at part load = steam mass flow ratio × p_1

Figure 7.31 *Mollier diagram for a steam turbine*

$$= 0.2 \times 20 = 4 \text{ bar}$$
$$= 3020 - 2350 = 670 \text{ kJ/kg}$$
$$= 0.7 \times 7.246 \times 670 \times 0.2 = 679.71 \text{ kW}$$

Coupling power = 679.71 – 200 = 479.71 kW

Example 7.11

The turbine rotor has a mean blade ring diameter of 500 mm and the

blade angles are equal. The nozzle angle is 20° and the steam leaves the nozzles with a velocity of 900 m/s. Assuming a blade friction factor of 0.85, determine the (a) the best blade angles, (b) the turbine speed in rpm, (c) the steam consumption rate in kg/h if the turbine generates 10 kW, and (d) the maximum blade efficiency.

[IES, 2006]

Solution

Given that $\alpha_1 = 20°$, $d_m = 0.5$ m, $v_{a1} = 900$ m/s, K = 0.85, $\beta = \beta_1 = \beta_2$ so that C = 1, $P = 10$ kW

The velocity diagrams are shown in Fig. 7.32.

For maximum blade efficiency, the speed ratio,

Figure 7.32 *Velocity diagrams for an impulse turbine*

$$v_w = v_{w1} + v_{w2}$$
$$= v_{r1} \cos \beta_1 + v_{r2} \cos \beta_2$$
$$= v_{r1} \cos \beta \,(1 + \mathrm{KC}) = 18.5\ v_{r1} \cos \beta$$
$$u = v_{a1} \cos \alpha_1 - v_{r1} \cos \beta$$
$$v_{r1} \cos \beta = 900 \cos 20° - 422.86 = 422.86 \text{ m/s}$$
$$v_w = 1.85 \times 422.86 = 782.25 \text{ m/s}$$

Power developed by turbine, P =

Mass flow rate, = 0.03023 kg/s or 108.83 kg/h

Maximum blade efficiency,

or $v_{r1} = 523.03$ m/s

or $\beta = 36°$

Figure 7.33 *Steam turbine blades erosion*

The steam turbine blades are subjected to high pressure and temperature. In the intermediate pressure stages, the steam is wet. Therefore, the material of blades should be able to withstand corrosion and erosion due to the presence of water particles. In addition to corrosion and erosion, blades are subjected to high centrifugal stresses. When the speed is high and the dryness fraction is less than 0.9, the effect of moisture is the most prominent. The most affected position is the back of the inlet edge of the blade, where either grooves are formed or even some portion breaks away. Due to centrifugal force, the water particles tend to concentrate in the outer annulus and their tip speed is greater than the

root speed. Hence, erosion effect is the most prominent on the tips, as shown in Fig. 7.33.

The following methods may be adopted to prevent erosion:

1. By raising the temperature of steam at inlet so that at the exit of the turbine, the dryness fraction does not fall below 0.9.
2. By adopting reheat cycle so that the dryness fraction at exit remains within limits.
3. By providing drainage belts on the turbine so that the water droplets which are on outer periphery, due to centrifugal force, are drained.
4. By providing a shield of a hard material on the leading edge of the turbine.

The most satisfactory solution to prolong the blade life is providing tungsten shield.

Example 7.12

In a De-Laval turbine, steam enters

the wheel through a nozzle with a velocity of 500 m/s and at an angle of 20° to the direction of motion of the blade. The blade speed is 200 m/s and the exit angle of the moving blade is 25°. Find the inlet angle of the moving blade, exit velocity of steam and its direction and work done per kg of steam.

Solution

Given that $v_{a1} = 500$ m/s; $\alpha_1 = 20°$; $u = 200$ m/s; $\beta_2 = 20°$, $v_{r2} = v_{r1}$.

Now let us draw the combined velocity triangles, as shown in Fig. 7.34, as explained below:

1. First, draw a horizontal line and cut off AB equal to 200 m/s, to some suitable scale, representing the blade speed, *u*.

2. Now at B, draw a line BC at an angle of 20° (nozzle angle, α_1) and cut off BC equal to 500 m/s to the same scale to represent the velocity of steam jet entering the blade (v_{a1}).
3. Join AC, which represents the relative velocity at inlet (v_{r1}).
4. At A, draw a line AD at an angle of 25° (exit angle of the moving blade, β_2). Now with A as centre, and radius equal to AC, draw an arc meeting the line through A at D.

Figure 7.34 *Velocity diagrams for De-Laval steam turbine*

5. Join BD, which represent the velocity of steam jet at outlet (v_{a2}).
6. From C and D, draw perpendiculars meeting the line AB produced at E and F respectively. CE and DF represent the velocity of flow at inlet (v_{f1}) and outlet (v_{f2}) respectively.

The following values are measured from the velocity diagram:

$$\beta_1 = 32°, \alpha_2 = 59°; v_{a2} = BD = 165 \text{ m/s}$$
$$v_{w1} = BE = 470 \text{ m/s and } v_{w2} = BF = 90 \text{ m/s}$$

By measurement from the velocity diagram, the inlet angle of the moving blade, $\beta_1 = 32°$

By measurement from the velocity diagram, the exit velocity of steam, $v_{a2} = 165$ m/s

By measurement from the velocity diagram, the direction of the exit steam, $\alpha_2 = 59°$

Work done per kg of steam = $\dot{m}$ $(v_{w1} + v_{w2})$

$$= 1\ (470 + 90) = 560 \text{ N-m/kg} \qquad \because (\dot{m} = 1 \text{ kg})$$

Example 7.13

The velocity of steam leaving the nozzles of an impulse turbine is 1200 m/s and the nozzle angle is 20°. The blade velocity is 375 m/s and the blade velocity coefficient is

0.75. Assuming no loss due to shock at inlet, calculate for mass flow of 0.5 kg/s and symmetrical blading: (*a*) blade inlet angle; (*b*) driving force on the wheel; (*c*) axial thrust on the wheel; and (*d*) power developed by the turbine.

Solution

Given that $v_{a1} = 1200$ m/s; $\alpha_1 = 20°$; $u = 375$ m/s; $K = v_{r2}/v_{r1} = 0.75$; $\dot{m} = 0.5$ kg/s; $\beta_1 = \beta_2$, for symmetrical blading.

Now draw the combined velocity triangle, as shown in Fig. 7.35, as explained below:

1. First, draw a horizontal line, and cut off AB equal to 375 m/s to some suitable scale representing the velocity of blade(u).

2. Now at B, draw a line BC at an angle of 20° (Nozzle angle, α_1) and cut off BC equal to 1200 m/s to the scale to represent the velocity of steam jet entering the blade (v_{a1}).

Figure 7.35 *Velocity diagrams for impulse turbine*

3. Join CA, which represents the relative velocity at inlet (v_{r1}). By measurement, we find that CA = v_{r1} = 860 m/s. Now cut off AX equal to $v_{r2} = v_{r1} \times K = 860 \times 0.75 = 645$ m/s to the scale to represent the relative velocity at exit (v_{r2}).
4. At A, draw a line AD at an angle β_2 equal to the angle β_1, for symmetrical blading. Now with A as centre, and radius equal to AX, draw an arc meeting the line through A at D, such that AD = v_{r2}.
5. Join BD, which represents the velocity of steam jet at outlet (v_{a2}).
6. From C and D, draw perpendiculars meeting the line AB produced at E and F respectively. CE and DF represents the velocity of flow at inlet (v_{f1}) and outlet (v_{f2}), respectively.

The following values are measured from the velocity diagram:

$$\beta_1 = 29°,\ v_{w1} = \text{BE} = 1130 \text{ m/s};\ v_{w2} = \text{BF} = 190 \text{ m/s}$$
$$v_{f1} = \text{CE} = 410 \text{ m/s and } v_{f2} = \text{DF} = 310 \text{ m/s.}$$

1. By measurement from the velocity diagram, the blade angle at inlet, $\beta_1 = 29°$
2. Driving force on the wheel, $F_x = m\,(v_{w1} + v_{w2}) = 0.5\,(1130 + 190) = 660$ N
3. Axial thrust on the wheel, $F_y = m\,(v_{f1} - v_{f2}) = 0.5\,(410 - 310) = 50$ N
4. Power developed by the turbine,

$P = \dot{m}\,(v_{w1} + v_{w2})\,u = 0.5\,(1130 + 190)\,375 = 247500$ W or 247.5 kW

Example 7.14

A single row impulse turbine receives 3 kg/s steam with a velocity of 425 m/s. The ratio of blade speed to jet speed is 0.4 and the stage output is 170 kW. If the internal losses due to disc friction etc., amount to 15 kW, determine the blade efficiency and the blade velocity coefficient. The nozzle angle is 16° and the blade exit angle is 17°.

Solution

Given that $\dot{m} = 3$ kg/s; $v_{a1} = 425$ m/

s; $u/v_{a1} = 0.4$; stage output = 170 kW; internal losses = 15 kW; $\alpha_1 = 16°$; $\beta_2 = 17°$.

Figure 7.36 *Velocity diagrams for single row impulse turbine*

1. Blade speed, $u = v_{a1} \times 0.4 = 425 \times 0.4 = 170$ m/s

Total power developed, P = Stage output + Internal losses

$$= 170 + 15 = 185 \text{ kW}$$

Let $v_{w1} + v_{w2}$ = Change in the velocity of whirl

Power developed, $P = \dot{m}\,(v_{w1} + v_{w2})\,u$

$$185 \times 10^3 = 3\,(v_{w1} + v_{w2})\,170$$

$\therefore v_{w1} + v_{w2} = 363$ m/s

Now, draw the combined velocity triangle, as shown in Fig. 7.36, and explained below:

1. First, draw a horizontal line and cut off AB equal to 170 m/s, to some suitable scale, to represent the blade speed (u).
2. Now, draw the inlet velocity triangle ABC on the base AB with $\alpha_1 = 16°$ and $v_{a1} = 425$ m/s, to the scale, chosen.
3. Similarly, draw the outlet velocity triangle ABD on the same base AB with $\beta_2 = 17°$ and $(v_{w1} + v_{w2}) = 363$ m/s to the scale.
4. From C and D, draw perpendiculars to meet the line AB at E and F. From the geometry of the figure, we

find that v_{w2} is in the opposite direction of v_{w1}. Therefore, $(v_{w1} - v_{w2}) = 363$ m/s.

By measurement from the velocity diagram, we find that

Relative velocity at inlet, $v_{r1} = 265$ m/s

and relative velocity at outlet, $v_{r2} = 130$ m/s

Blading efficiency,

2. Blade velocity coefficient,

Example 7.15

In one stage of a reaction steam turbine, both the fixed and moving blades have inlet and outlet blade tip angles of 35° and 20°, respectively. The mean blade speed is 80 m/s and the steam consumption is 22500 kg per hour. Determine the power developed in the pair, if the isentropic heat drop for the pair is

23.5 kJ per kg.

Figure 7.37 *Velocity diagrams for reaction turbine*

Solution

Given that $\beta_1 = \alpha_2 = 35°$; $\beta_2 = \alpha_1 = 20°$; $u = 80$ m/s; $\dot{m} = 22500$ kg/h $= 6.25$ kg/s; $\Delta h = 23.5$ kJ/kg.

Now, let us draw the combined velocity triangle, as shown in Fig. 7.37 and explained below:

1. First, draw a horizontal line and cut off AB equal to 80 m/s (u) to some suitable scale.
2. Now at B, draw in line BC at an angle $\alpha_1 = 20°$ with AB. Similarly, at A draw a line AC at an angle $\beta_1 = 35°$ with BA meeting the first line at C.
3. At A, draw a line AD at angle $\beta_2 = 20°$ (because $\beta_2 = \alpha_1$) with AB. Similarly, at B draw a line BD at an angle $\alpha_2 = 35°$ (because $\alpha_2 = \beta_1$) with AB meeting the first line at D.
4. From C and D, draw perpendiculars meeting the line AB produced at E and F.

By measurement, we find that the

change in the velocity of whirl,

$$\Delta w = v_{w1} + v_{w2} = 235 \text{ m/s}$$

Power developed in the pair,

$$P = \dot{m}\,(v_{w1} + v_{w2})\,u = 6.25 \times 235 \times 80 = 117500 \text{ W} = 117.5 \text{ kW}$$

Example 7.16

A 50% reaction turbine with a mean blade diameter of 1 m runs at a speed of 50 rps. The blades are designed with exit angles of 50° and inlet angles of 30°. If the turbine is supplied with steam at the rate of 20 kg/s and gross efficiency is 85%, determine the following: (*a*) power output of the stage; (*b*) specific enthalpy drop in the stage; and (*c*)

percentage increase in relative velocity in the moving blades due to steam expansion.

Solution

Given that d_m = 1 m; N = 50 rps; $\beta_1 = \alpha_2 = 50°$; $\beta_2 = \alpha_1 = 30°$; m = 20 kg/s; η_s = 85%

1. Blade velocity, $u = \pi\, d_m\, N = \pi \times 1 \times 50 = 157$ m/s

   Now let us draw the combined velocity triangle, as shown in Fig. 7.38, and explained below:

   1. First, draw a horizontal line and cut off AB equal to 157 m/s, to some suitable scale, to represent the blade velocity u.
   2. Now, draw the inlet velocity triangle ABC on the base AB with $\alpha_1 = 30°$ and $\beta_1 = 50°$.
   3. Similarly, draw the outlet velocity triangle ABD on the same base AB with $\beta_2 = 30°$ and $\alpha_2 = 50°$.
   4. From C and D, draw perpendiculars to meet the line AB produced at E and F.

   By measurement from the velocity triangle, we find that change in the velocity of whirl.

$$v_{w_1} + v_{w_2} = \text{EF} = 450 \text{ m/s}$$

Relative velocity at inlet, v_{r1} = CA = 230 m/s

Figure 7.38 *Velocity diagrams for symmetrical blades reaction turbine*

and relative velocity at outlet, v_{r2} = DA = 350 m/s

Power output of the stage,

$$P = \dot{m}\,(v_{w1} + v_{w2})\,u = 20 \times 450 \times 157 = 1413000 \text{ W} = 1413 \text{ kW.}$$

2. Let Δh = specific enthalpy drop in the stage.

We know that stage efficiency (η_s);

3. Increase in the relative velocity in the moving blades due to steam expansion.

Example 7.17

At a stage of a 50% reaction turbine, the rotor diameter is 1.4 m and speed ratio 0.7. If the blade outlet angle is 20° and the rotor speed 3000 rpm, find the blade inlet angle and diagram efficiency.

Find the percentage increase in diagram efficiency and rotor speed, if the turbine is designed to run at the best theoretical speed.

Solution

Given that $D = 1.4$ m; $\rho = u/v_{a1} = 0.7$; $\alpha_1 = \beta_2 = 20°$; $N = 3000$ rpm.

Blade velocity,

and velocity of steam at inlet to the blade.

$$v_{a1} = u/0.7 = 220/0.7 = 314.3 \text{ m/s}$$

Now draw the combined velocity triangle, as shown in Fig. 7.39, as explained below:

1. First, draw a horizontal line and cut off AB equal to 220 m/s, to some suitable scale, to represent the blade velocity (u).

Figure 7.39 *Velocity diagrams for reaction turbine with R_d = 50%*

2. Now, draw the inlet velocity triangle ABD on the same base AB with $\alpha_1 = 20°$ and $v_{a1} = 314.3$ m/s, to the scale.
3. Similarly, draw the outlet velocity triangle ABD on the same base AB with $\beta_2 = 20°$ and $v_{r2} = 314.3$ m/s to the scale.
4. From C and D, draw perpendiculars to meet the line AB produced at E and F.

By measurement from velocity diagram, we find that the blade inlet angle.

$$\beta_1 = 55°$$

By measurement from velocity diagram, we find that velocity of steam at outlet,

$$v_{a2} = 130 \text{ m/s}$$

Diagram efficiency,

Maximum efficiency of the turbine,

∴ Percentage increase in diagram efficiency = = 0.131 or 13.1%

Let N_1 = Maximum rotor speed.

For best theoretical speed (or in other words, for maximum efficiency), the blade velocity,

$$u = v_{a1} \cos \alpha_1 = 314.3 \cos 20° = 314.3 \times 0.9397 = 295.3 \text{ m/s}$$

Blade velocity (u);

∴ Percentage increase in rotor speed = = 0.348 or 34.8%

Example 7.18

One stage of an impulse turbine consists of a converging nozzle ring and one ring of moving blades. The nozzles are inclined at 22° to the blades whose tip angles are both 35°. If the velocity of steam at exit from the nozzle is 660 m/s, find the blade speed so that the steam shall pass on without shock. Find the diagram efficiency neglecting losses if the blades are run at this speed.

[IES, 1992]

Solution

Given that $\alpha_1 = 22°$, $\beta_1 = \beta_2 = 35°$, $v_{a1} = 660$ m/s

The velocity triangles for impulse

turbine are shown in Fig. 7.40.

Maximum blade efficiency,

For $\beta_1 = \beta_2$, $C = 1$ and for no friction on the blades, $K = 1$.

$\therefore (\eta_b)_{max} = \cos^2 \alpha_1 = \cos^2 22° =$ 0.8597 or 85.97%

Speed ratio, $\rho =$

Blade speed, $u = 660 \times = 305.97$ m/s.

Figure 7.40 *Velocity triangles for one-stage impulse steam turbine*

Example 7.19

Steam expands in a turbine from 40

bar, 450°C to 0.1 bar isentropically. Assuming ideal conditions, determine the mean diameter of the wheel if the turbine were of (a) single impulse stage, (b) single 50% reaction stage, (c) four pressure (or Rateau) stages, (d) one two-row Curtis stage, and (e) four 50% reaction stages. Take the nozzle angle as 15° and speed 3000 rpm.

Solution

Given that $p_1 = 40$ bar, $t_1 = 450°C$, $p_2 = 0.9$ bar, $\alpha_1 = 15°$, N = 3000 rpm

At 40 bar, 450°C, from steam tables, we have

$$h_1 = 3030.2 \text{ kJ/kg}, \; s_1 = 6.9362 \text{ kJ/kg.K}$$

At 0.1 bar: $s_{f2} = 0.6492$ kJ/kg.K, $s_{fg2} = 7.5010$ kJ/kg.K,

$$h_{f2} = 191.81 \text{ kJ/kg}, \; h_{fg2} = 2392.8 \text{ kJ/kg}$$

Now for isentropic flow, $s_1 = s_2 = s_{f2} + x_2 \, s_{fg2}$

or $6.9362 = 0.6492 + x_2 \times 7.5010$

or $x_2 = 0.838$

$$h_2 = h_{f2} + x_2 \, h_{fg2} = 191.81 + 0.838 \times 2392.8 = 2196.98 \text{ kJ/kg}$$

$$\Delta h = h_1 - h_2 = 3030.2 - 2196.98 = 833.22 \text{ kJ/kg}$$

1.

2.
3.
4.

Example 7.20

Steam at 20 bar, 500°C expands in a

steam turbine to 0.01 bar. There are four stages in the turbine and the total enthalpy drop is divided equally among the stages. The stage efficiency is 75% and it is the same in all the stages. Calculate the interstage pressures, the reheat factor, and the turbine internal efficiency.

Solution

Given that $p_1 = 20$ bar, $t_1 = 500°C$, $p_2 = 0.09$ bar, $\eta_s = 0.75$

At 20 bar, 500°C, $h_1 = 3467.6$ kJ/kg, $s_1 = 7.4316$ kJ/kg.K

At 0.1 bar, $h_{f6} = 191.81$ kJ/kg, $h_{fg6} = 2392.8$ kJ/kg, $s_{f6} = 0.6492$ kJ/kg.K,

$s_{fg6} = 7.5010$ kJ/kg.K

Now $s_1 = s_6 = s_{f6} + x_6\ s_{fg6}$

or $7.4316 = 0.6491 + x_6 \times 7.5010$

or $x_6 = 0.9042$

$$h_6 = h_{f6} + x_6\ h_{fg6} = 191.81 + 0.9042 \times 2392.8 = 2355.4 \text{ kJ/kg}$$
$$h_1 - h_6 = 3467.6 - 2355.4 = 1112.2 \text{ kJ/kg}$$

The Mollier diagram is shown in Fig. 7.41.

The inter stage pressures read from Mollier chart shown in Fig. 7.41 are:

Reheat factor =

Figure 7.41 *h-s diagram*

Turbine internal efficiency, η_{it} =

Example 7.21

Find the maximum blade efficiency and corresponding blade angles for a single row impulse steam turbine assuming equiangular blades, when the nozzle angle $\alpha_1 = 20°$ and blade velocity coefficient $K = 0.9$.

If the blade efficiency is 85% of the maximum value, what are the possible blade speed ratios for the same nozzle angle α_1, blade velocity coefficient, K and equiangular blades? Find the corresponding blade angles.

[IAS, 2003]

Solution

Given that $\alpha_1 = 20°$, $K = 0.9$

Maximum blade efficiency, $(\eta_b)_{\text{max}}$ $= (1 + \text{KC})$

where $K =$ and $C =$

Figure 7.42 *Velocity diagrams for impulse turbine*

or $\beta_1 = 36°$

Let the blades be symmetrical so that $\beta_2 = \beta_1 = 36°$ and C = 1

The velocity diagrams at inlet and outlet of a moving blade are shown in Fig. 7.42.

Example 7.22

Steam expands in a steam turbine isentropically from inlet to exhaust having an enthalpy drop = 12000 kJ/kg. Assuming ideal conditions, determine the mean diameter of the wheel if the turbine were of:

1. Single impulse stage
2. Single 50% reaction stage
3. One two-row Curtis stage
4. Ten 50% reaction stages

Take the nozzle angle as 18° and blade speed as 4000 rpm

[IAS, 2002]

Solution

Given that $\alpha_1 = 18°$, $N = 4000$ rpm, $\Delta h = 12000$ kJ/kg,

1. Single impulse stage turbine:

   For ideal conditions,

   Speed ratio, = 0.47553

   or $u = 0.47553 \times 4898.98 = 2329.6$ m/s

   or

2. Single 50% reaction stage:

   or $u = 4898.98 \times \cos 18° = 4659.2$ m/s

3. One two-row Curtis stage:
4. Ten 50% reaction stage

Example 7.23

The velocity of steam entering a simple impulse turbine is 1000 m/s and the nozzle angle is 20°. The mean peripheral velocity of blades is 400 m/s. The blades are asymmetrical. If the steam is to

enter the blades without shock, what will be the blade angles?

Neglecting the friction effects on the blades, calculate the tangential force on the blades and the diagram power for a mass flow of 0.75 kg/s. Calculate the axial thrust and diagram efficiency.

[IAS, 2001]

Solution

Given that v_{a1} = 1000 m/s, α_1 = 20°, u = 400 m/s, $v_{r1} = v_{r2}$, $\dot{m}$ = 0.75 kg/s. For symmetrical blades,

$$\beta_2 = \alpha_1 = 20°, \alpha_2 = \beta_2$$

Steam is to enter the blades without shock.

The velocity diagrams are shown in Fig. 7.43.

$$v_{w1} = v_{a1} \cos \alpha_1 = 1000 \cos 20° = 939.7 \text{ m/s}$$
$$v_{f1} = v_{a1} \sin \alpha_1 = 1000 \sin 20° = 342.02 \text{ m/s}$$
$$= 400^2 + 1000^2 - 2 \times 400 \times 1000 \times \cos 20°$$
$$= 408,245.9$$

or $v_{r1} = 638.94$ m/s

or $\beta_1 = 32.36°$

$\therefore \alpha_2 = 32.36°$

$$v_{r2} = v_{r1} = 638.94 \text{ m/s}$$
$$v_{w2} = v_{r2} \cos \beta_2 - u = 638.94 \cos 20° - 400 = 200.4 \text{ m/s}$$

Tangential force on the blades, $F_t = \dot{m}\,(v_{w1} + v_{w2}) = 0.75\,(939.7 + 200.4) = 855.07$ N

Diagram power = = 342.03 kW

Figure 7.43 *Velocity diagrams for simple impulse turbine*

Diagram efficiency = 0.9121 or 91.21%

Axial thrust, $F_a = \dot{m}\,(v_{f1} - v_{f2})$ = 0.75 (342.02 – 218.53) = 92.6 N

Summary for Quick Revision

1. Steam turbines are of three types: impulse turbines, impulse reaction turbines, and reaction turbines.
2. Compounding of steam turbines is a method employed for reducing the rotational speed of the impulse turbine to practical limits by using more than one stage.
3. Impulse steam turbines can be compounded by velocity compounding, pressure compounding, and mixed velocity and pressure compounding.
4. Impulse turbine:
    1. Power developed, P = kW
    2. Diagram or blade efficiency, η_b =
    3. Gross or stage efficiency, η_s =
    4. Nozzle efficiency, η_n =
    5. Axial thrust, F_a =
    6. Energy converted into heat by blade friction
    7. For maximum blade efficiency: speed ratio,

    where

    where β_1 = inlet angle of moving blade, β_2 = exit angle of moving blade, α_1 = outlet angle of fixed blade

For symmetrical blades, $\beta_1 = \beta_2$ and $C = 1$

$$(\eta_b)_{max} = \cos^2 \alpha_1$$

Work done, $w = (v_{a1} \cos \alpha_1 - u)(1 + KC)u$

$$w_{max} = 2u^2 \text{ for } K = 1 \text{ and } C = 1$$

5. Velocity compounded impulse turbine:
   1. For symmetric blades and no friction loss, i.e. $\beta_1 = \beta_2$ and $v_{r1} = v_{r2}$

      Total work done, $w_t = 4u(v_{a1} \cos \alpha_1 - 2u)$

   2. Blade efficiency, $\eta_b = 8\rho(\cos \alpha_1 - 2\rho)$

      For maximum blade efficiency,

      and $(\eta_b)_{max} = \cos^2 \alpha_1$

      $$(w_t)_{max} = 8u^2$$

   3. If n = number of rotating blade rows in series, then

      and work done in the last row =

6. Reaction turbine:
   1. Degree of reaction, R_d =
   2. For $R_d = 0.5$,

      $$u = v_f (\cot \beta_2 - \cot \alpha_2) = v_f (\cot \alpha_1 - \cot \beta_1)$$

      $\therefore \beta_1 = \alpha_2$ and $\beta_2 = \alpha_1$

      Also $v_{r2} = v_{a1}$

   3. Work done per kg of steam, $w = [2\rho \cos \alpha_1 - \rho^2]$
   4. Blade efficiency, η_b =
   5. For maximum blade efficiency, $\rho = \cos \alpha_1$
   6. If n = number of blades, and d = mean diameter of turbine wheel then pitch of blades, p =
   7. Volume flow rate, $\dot{m}\, v_s = (\pi d - nt_1)h_1 v_{f1} = (\pi d - nt_2)$

$h_2 v_{f2}$

For $v_{f1} = v_{f2} = v_f$, $t_1 = t_2 = t$ and $h_1 = h_2 = h$

$$\dot{m}\, v_s = (\pi d - nt)\, h v_f$$

Height of blades, $h =$

8. Reheat factor, $R_f =$
9. Turbine internal efficiency, $\eta_i = \eta_s \times R_f$

where η_s = stage efficiency

7. Turbine efficiencies:
    1. $\eta_b =$
    2. $\eta_s =$
    3. $\eta_i =$
    4. Overall efficiency, $\eta_{\text{overall}} =$
    5. Efficiency ratio or net efficiency η_{net}, =
8. Governing of steam turbines:
    1. Throttle governing
    2. Nozzle control governing, and
    3. By-pass governing.

Multiple-choice Questions

1. In steam turbine terminology, diaphragm refers to
    1. The separating wall between rotors carrying nozzles
    2. The ring of guide blades between rotors
    3. A partition between low and high pressure dies
    4. The flange connecting the turbine exit to the condenser
2. A three-stage Rateau turbine is designed in such a manner that the first two stages develop equal power with identical velocity diagram, whereas the third one develops more power with higher blade speed. In such a multistage turbine, the blade ring diameter
    1. is the same for all the three stages
    2. gradually increases from the first to the third stage

    3. of the third stage is greater than that of the first two stages
    4. of the third stage is less than that of the first two stages
3. In a De Laval nozzle expanding superheated steam from 10 bar to 0.1 bar, the pressure at the minimum cross-section will be
    1. 3.3 bar
    2. 5.46 bar
    3. 8.2 bar
    4. 9.9 bar
4. A single-stage impulse turbine with a diameter of 120 cm runs at 300 rpm. If the blade speed ratio is 0.42, then, the inlet velocity of steam will be
    1. 79 m/s
    2. 188 m/s
    3. 450 m/s
    4. 900 m/s
5. In an ideal impulse turbine, the
    1. absolute velocity at the inlet of moving is equal to that at the outlet
    2. relative velocity at the inlet of the moving blade is equal to that at the outlet
    3. axial velocity at the inlet is equal to that at the outlet
    4. whirl velocity at the inlet is equal to the outlet
6. For a Parson's reaction turbine, if α_1 and α_2 are fixed angles at inlet and exit, respectively, and β_1 and β_2 are the moving blade angles at entrance and exit, respectively, then
    1. $\alpha_1 = \alpha_2$ and $\beta_1 = \beta_2$
    2. $\alpha_1 = \beta_1$ and $\alpha_2 = \beta_2$
    3. $\alpha_1 < \beta_1$ and $\alpha_2 > \beta_2$
    4. $\alpha_1 < \beta_2$ and $\beta_1 > \alpha_2$
7. The isentropic enthalpy drop in moving blade is two-thirds of the isentropic enthalpy drop in pin fixed blades of a turbine. The degree of reaction will be
    1. 0.4
    2. 0.6
    3. 0.66
    4. 1.66
8. The efficiency of the nozzle-governed turbines is affected mainly by losses due to
    1. partial admission
    2. throttling
    3. inter-stage pressure drop

    4. condensation in last stages
9. The main aim of compounding steam turbine is to
    1. improve efficiency
    2. reduce steam consumption
    3. reduce motor speed
    4. reduce turbine size
10. A throttle-governed steam develops 20 kW with 281 kg/h of steam and 50 kW with 521 kg/h of steam. The steam consumption in kg/hr when developing 15 kW will be nearly
    1. 150 kg/hr
    2. 156 kg/hr
    3. 241 kg/hr
    4. 290 kg/hr
11. Governing of steam turbines can be done by the following:

        1. Nozzle control
        2. Throttle control
        3. Providing additional value and passage

    The correct answer will be

    1. 1, 2, and 3
    2. 1 and 2 only
    3. 2 and 3 only
    4. 1 and 3 only
12. What is the cause of reheat factor in a steam turbine?
    1. Reheating
    2. Superheating
    3. Supersaturation
    4. Blade friction
13. In a steam power plant, feed water heater is a heat exchanger to preheat feed water by
    1. live steam from steam generator
    2. hot flue gases coming out of the boiler furnace
    3. hot air from air preheater
    4. extracting steam from turbine
14. Match List I (turbines) with List II (classification) and select correct answer using the codes given below the List:

Codes:

A B C D

1. 3 2 1 4
2. 2 3 4 1
3. 2 3 1 4
4. 3 2 4 1

15. The outward radial flow turbine in which there are two rotors rotating in opposite directions is known an
    1. 50% reaction radial turbine
    2. Cantilever turbine
    3. Ljungstrom turbine
    4. Pass-out turbine
16. In reaction turbines, with reduction of inlet pressure
    1. the blade heights increase as the specific volume of steam decreases
    2. the blade heights increase as the specific volume of steam increases
    3. the blade heights decrease as the specific volume of steam increases
    4. the blade heights decrease as the specific volume of steam decreases
17. Which of the following statements are correct?

    1. Impulse turbine rotor blades are thick at the centre
    2. Rateau turbine is more efficient than Curtis turbine
    3. Blade velocity coefficient for an impulse turbine is the order of 60%

    Select the correct answer using the codes given below:

    1. 1, 2, and 3
    2. 1 and 2
    3. 1 and 3
    4. 2 and 3
18. Given that α_1 = nozzle angle, n = number of rows of moving

blades in a velocity compounded, impulse turbine, the optimum blade speed ratio is

1. 2 cos $a_1 n$
2.
3.
4.

19. In a Parson's turbine stage, blade velocity is 320 m/s at the mean radius and the rotor blade exit angle is 30°. For minimum kinetic energy of the steam leaving the stage, the steam velocity at the exit of the stator will be
    1.
    2. 640 m/s
    3.
    4.
20. For a free vortex design of blade in the rotor of a reaction axial turbine, the specific work along the blade height is
    1. higher at the blade hub and lower at the blade tip
    2. constant from hub to tip
    3. lower at the hub and tip but different from the mean section
    4. the same at the hub and tip but different from the mean section
21. Which of the following sketches represents an impulse turbine blade?
    1.
    2.
    3.
    4.
22. The graph shown in Fig. 7.44 represents the variation of absolute velocity of steam along the length of a steam turbine.

Figure 7.44

The turbine in question is

1. Curtis turbine
2. De-Laval turbine
3. Radial turbine
4. Parson's turbine

23. In a simple impulse turbine, the nozzle angle at the entrance is 30°. What will be the blade-speed ratio for maximum diagram efficiency?
    1. 0.433

2. 0.25
3. 0.5
4. 0.75

24. A reaction turbine stage has angles α_1, β_1, β_2 as nozzle angle, inlet blade and outlet blade angle, respectively. The expression for maximum efficiency of the turbine is given by
    1.
    2.
    3.
    4.
25. The correct sequence of the given steam turbines in the ascending order of efficiency at their design point is
    1. Rateau, De- Laval, Parson's, Curtis
    2. Curtis, De- Laval, Rateau, Parson's
    3. De -Laval, Curtis, Rateau, Parson's
    4. Parson's, Curtis, Rateau, De- Laval
26. Consider the following statements regarding the nozzle governing of steam turbines:

    1. Working nozzles receive steam at full pressure
    2. High efficiency is maintained at all loads
    3. Stage efficiency suffers due to partial admission
    4. In practice, each nozzle of the first stage is governed individually

    Of these statements,

    1. 1, 2, and 3 are correct
    2. 2, 3, and 4 are correct
    3. 1, 3, and 4 correct
    4. 1, 2, and 4 are correct
27. Among other things, the poor part-load performance of De Laval turbines is due to the
    1. formation of shock waves in the nozzle
    2. formation of expansion waves at the nozzle
    3. turbulent mixing at the nozzle exit
    4. increase in profile losses in the rotor
28. List I gives the various velocities in the velocity diagrams of a two-stage impulse turbine. List II gives the blade angles. Match the velocity from List I with the angle in List II and select the correct answer using the codes given below the lists:

Codes:

	A	B	C	D
1.	1	2	4	3
2.	2	1	4	3
3.	2	1	3	4
4.	1	2	3	4

29. Which of the following relationship between angles of fixed blades and moving blades corresponds to that of Parson's turbine?
    1. $\alpha_1 = \alpha_2$
    2. $\alpha_1 = \beta_2$
    3. $\alpha_2 = \beta_2$
    4. $\beta_1 = \beta_2$
30. The following data refer to an axial flow turbine stage:

    Relative velocity of steam at inlet to the rotor = 79.0 m/s., Relative velocity at the rotor exit = 152 m/s. What is the approximate degree of reaction?

    1. 0.9
    2. 0.8
    3. 0.7
    4. 0.6
31. The clearance flow between the blade tips and casing of a steam turbine is
    1. greater in the reaction turbine than in the impulse type
    2. greater in the impulse turbine than in the reaction type
    3. independent of type of the turbine
    4. independent of the size of the turbine
32. Running speeds of steam turbine can be brought down to practical limits

    1. By using heavy flywheel
    2. By using a quick response governor
    3. By compounding

4. By reducing fuel feed to the furnace

1. Only 3
2. 1, 2, 3, and 4
3. 1, 2, and 4
4. 2 and 3

33. The net result of pressure-velocity compounding of steam turbine is
    1. less number of stages
    2. large turbine for a given pressure drop
    3. shorter turbine for a given pressure drop
    4. lower friction loss
34. Given, v_b = Blade speed

    v_{a1} = Absolute velocity of steam entering the blade, α_1= Nozzle angle

    The efficiency of an impulse turbine is maximum when

    1. $v_b = 0.5\, v_{a1} \cos \alpha_1$
    2. $v_b = v_{a1} \cos \alpha_1$
    3. $v_b = 0.5\, v^2{}_{a1} \cos \alpha_1$
    4. $v_b = v^2{}_{a1} \cos \alpha_1$
35. An impulse turbine produces 50 kW of power when the blade mean speed is 400m/s. What is the rate of change of momentum tangential to the rotor?
    1. 200 N
    2. 175 N
    3. 150 N
    4. 125 N
36. At a particular section of a reaction turbine, the diameter of the blade is 1.8 m, the velocity of flow of steam is 49 m/s and the quantity of steam flow is 5.4 m^3/s. The blade height at this section will be approximately:
    1. 4 cm
    2. 2 cm
    3. 1 cm
    4. 0.5 cm
37. Consider the following statements:

If steam is reheated during the expansion through turbine stages,

1. Erosion of blade will decrease
2. The overall pressure ratio will increase
3. The total heat drop will increase

Of these statements,

1. 1, 2, and 3 are correct
2. 1 and 2 are correct
3. 2 and 3 are correct
4. 1 and 3 are correct

38. In an impulse-reaction turbine stage, the heat drop in fixed and moving blades are 15 kJ/kg and 30 kJ/kg respectively. The degree of reaction for this stage will be
    1. 1/3
    2. 1/2
    3. 2/3
    4. 3/4
39. If 'D' is the diameter of the turbine wheel and 'U' is its peripheral velocity, then the disc friction loss will be proportion to
    1. $(DU)^3$
    2. D^2U^3
    3. D^3U^2
    4. DU^4
40. In a two-row Curtis stage with symmetrical blading.
    1. work done by both rows of moving blades are equal
    2. work done by the first row of moving blades is double of the work done by second row of moving blades
    3. work done by the first row of moving blades is three times the work done by second row of moving blades
    4. work done by the first row of moving blades is four times the wok done by the second row of moving blades
41. The compounding of steam turbines is done to
    1. improve efficiency
    2. reduce turbine speed
    3. increase blade speed ratio
    4. refuse axial thrust

42. The expression for the maximum efficiency of a Parson's turbine is (α_1 is the angle made by absolute velocity an inlet)
    1.
    2.
    3.
    4.
43. Consider the following statements regarding effects of heating of steam in a steam turbine:

    1. It increases the specific output of the turbine
    2. It decreases the cycle efficiency
    3. It increases blade erosion
    4. It improves the quality of exit steam.

    Which of these statements are correct?

    1. 1, 2, and 3
    2. 2 and 3
    3. 3 and 4
    4. 1 and 4
44. Match List I (Different turbine stages) with List II (Turbines) and select the correct answer using the codes given below the lists:

    Codes:

    A B C D

    1. 5 1 2 3
    2. 5 3 2 1
    3. 2 3 4 1
    4. 3 1 4 2
45. Partial admission steams turbine refers to the situation where the
    1. steam is admitted partially into the blades through nozzles
    2. nozzle occupy the complete circumference leading into the blade annulus

3. nozzle do not occupy the complete circumference leading into the blade annulus
4. steam is admitted partially into the blades directly

46. Consider the following statements regarding a 100% reaction turbine:

    1. Change in absolute velocity of steam across the moving blades is zero.
    2. Change in absolute velocity of steam across the moving blades is negative.
    3. Enthalpy drop in fixed blades is zero.

    Which of these statements is/are correct?

    1. Only 1
    2. Only 2
    3. 2 and 3
    4. 1 and 3
47. Which one of the following pairs is not correctly matched?
    1. Internal efficiency of steam turbine: Product of stage efficiency and reheat factor
    2. Stage efficiency of a turbine: Ratio of adiabatic heat drop to the isentropic heat drop per stage
    3. Dryness fraction of steam within a stage: Decreases due to reheating
    4. Steam condensation during expansion through the turbine: Enhances blade erosion
48. Consider the following characteristics:

    1. High steam and blade velocities
    2. Low steam and blade velocities
    3. Low speeds of rotation
    4. High carry-over loss

    Which of these characteristics are possessed by a simple impulse turbine?

    1. 1 and 2
    2. 2 and 3
    3. 1 and 4

4. 3 and 4

49. Velocity triangle for a reaction turbine stage is shown in the given Fig. 7.45 (AB = v_{a1} = absolute velocity at rotor blade inlet; CB = v_{r1} = relative velocity at rotor blade inlet; CE = v_{r2} = relative velocity at rotor blade exit and CD = CB)

Figure 7.45

The ratio of reaction force to impulse force is

1. CE/CB
2. CD/CE
3. DE/BD
4. AE/AB

50. Consider the following statements:

1. Throttle governing improves quality of steam in the last few stages
2. Internal efficiency of steam is not seriously affected by throttle governing
3. Throttle governing is better than nozzle governing

Which of these statements are correct?

1. 1, 2, and 3
2. 1 and 3
3. 2 and 3
4. 1 and 2

51. Which one of the following statements is correct?
    1. Reheat factor is zero if efficiency of the turbine is close to unity
    2. Lower the efficiency, higher will be the reheat factor
    3. Reheat factor is independent of steam conditions at turbine inlet
    4. Availability of reheat is higher at low pressure end

52. For maximum blade efficiency of a single-stage impulse turbine, the blade speed ratio, (α_1 is, the angle made by absolute velocity at inlet) should be
    1. cos $2\alpha_1$
    2.
    3.

4.

53. Figure 7.46 shows the variation of certain steam parameter in case of a simple impulse turbine. The curve A-B-C represents the variation of

Figure 7.46

1. pressure in nozzle and blades
2. velocity in nozzle and blades
3. temperature in nozzle and blades
4. enthalpy in nozzle and blades

54. For a reaction turbine with degree of reaction equal to 50%, (v_{a1} is the absolute steam velocity an inlet and α_1 is the angle made by it to the tangent on the wheel) the efficiency is maximum when the blade speed is equal to

1.
2. $2v_{a1} \cos \alpha_1$
3. $v_{a1} \cos^2 \alpha_1$
4. $v_{a1} \cos \alpha_1$

55. Which one of the following is the correct statement? The degree of reaction of an impulse turbine

1. is less than zero
2. is greater than zero
3. is equal to zero
4. increases with steam velocity at the inlet

56. Why is compounding of steam turbines done?

1. To improve efficiency
2. To reduce the speed of rotor
3. To reduce exit losses
4. To increase the turbine output

57. A four-row velocity compounded steam turbine develops 6400 kW. What is the power developed by the last row?

1. 200 kW
2. 400 kW
3. 800 kW
4. 1600 kW

58. Which of the following is used to bring down the speed of an impulse steam turbine to practical limits?

1. A centrifugal governor
2. Compounding of the turbine
3. A large flywheel
4. A gear box

59. Consider the following for a steam turbine power plant:

1. Reduction in blade erosion

2. Increase in turbine speed
3. Increase in specific output
4. Increase in cycle efficiency

Which of the above occur/occurs due to reheating of steam?

1. Only 1
2. 1 and 2
3. 1, 3, and 4
4. 2 and 3

60. Ratio of enthalpy drop in moving blades to the total enthalpy drop in the fixed and moving blades is called
    1. reheat factor
    2. blade efficiency
    3. degree of reaction
    4. internal efficiency
61. Employing superheated steam in turbines leads to
    1. increase in erosion of blading
    2. decrease in erosion of blading
    3. no erosion in blading
    4. no change in erosion of blading
62. Steam enters a De laval steam turbine with an inlet velocity of 30 m/s and leaves with an outlet velocity of 10 m/s. The work done by 1 kg of steam is
    1. 400 Nm
    2. 600 Nm
    3. 800 Nm
    4. 1200 Nm
63. In a 50% reaction stage, absolute velocity angle at inlet is 45°, mean peripheral speed is 75 m/s and the absolute velocity at the exit is axial. The stage specific work is
    1. 2500 m^2/s^2
    2. 3270 m^2/s^2
    3. 4375 m^2/s^2
    4. 5625 m^2/s^2
64. Consider the following statements:

Steam turbines are suitable for use as prime movers for large steam power plants because

1. a single steam turbine can be designed for a capacity of 1000 MW or more
2. much higher speed may be possible as compared to a reciprocating engine
3. they are more compact when compared to a gas turbine power plant
4. the maintenance cost and running cost may not increase with years of service

Which of these statements are correct?

1. 1, 2 and 3
2. 2, 3 and 4
3. 1, 2 and 4
4. 1, 3 and 4

65. Expansion line EFG of a 2-stage steam turbine is shown in Fig. 7.47 (h-s diagram). The reheat factor for this turbine is

Figure 7.47

1. 1.08
2. 0.7
3. 0.648
4. 1.43

66. Which of the following is the features of pressure-compounding (Rateau staging)?
    1. Low efficiency at low rotational speed
    2. High efficiency with low fluid velocities
    3. High efficiency with high fluid velocities
    4. Low efficiency at high rotational speed
67. In Parson’s reaction turbines, the velocity diagram triangles at the inlet and outlet are
    1. asymmetrical
    2. isosceles
    3. right-angled
    4. congruent
68. In Parson’s turbine if α_1 is nozzle angle, then what is the maximum efficiency of the turbine?
    1.
    2.
    3.
    4.
69. What is the value of the reheat factor in multistage turbine?

1. 1.03 to 1.04
2. 1.10 to 1.20
3. 0.90 to 1.00
4. 1.20 to 1.25

70. In which one of the following steam turbines, is steam taken from various points along the turbine, solely for feed-water heating?
    1. Extraction turbine
    2. Bleeder turbine
    3. Regenerative turbine
    4. Reheat turbine
71. Velocity diagram shown in Fig. 7.48 is for an impulse turbine stage.

Figure 7.48

What are the tangential force and axial thrust per kg/s of steam, respectively?

    1. 450 N, 8 N
    2. 560 N, 8 N
    3. 680 N, 4 N
    4. 910 N, 4 N
72. Consider the following statements in respect of impulse steam turbines:
    1. Blade passages are of constant cross-sectional area.
    2. Partial admission of steam is permissible.
    3. Axial thrust is only due to change in flow velocity of steam an inlet and outlet of moving blade.

Which of the statements given above are correct?

    1. 1, 2, and 3
    2. Only 1 and 2
    3. Only 2 and 3
    4. Only 1 and 3
73. In an axial flow impulse turbine, energy transfer takes place due to
    1. change in relative kinetic energy

2. change in absolute kinetic energy
3. change in pressure energy
4. change in energy because of centrifugal force

74. In a reaction turbine the enthalpy drop in a stage is 60 kJ/kg. The enthalpy drop in the moving blades is 32 kJ/kg. What is the degree of reaction?
    1. 0.533
    2. 0.284
    3. 0.466
    4. 1.875
75. An emergency governor of a steam turbine trips the turbine when

    1. Shaft exceeds 100% of its rated speed
    2. Condenser becomes hot due to inadequate cooling water circulation
    3. Lubrication system fails
    4. Balancing of turbine is not proper

    Select the correct answer form the codes given below:

    Codes:

    1. 1, 2, and 3
    2. 2, 3 and 4
    3. 3, 4, and 1
    4. 4, 1, and 2
76. Steam enters the rotor of a reaction turbine with an absolute velocity of 236 m/s and relative velocity of 132 m/s. It leaves the rotor with a relative velocity of 2.32 m/s and absolute velocity of 136 m/s. The specific work output is
    1. 35.8 kW
    2. 40.1 kW
    3. 43.8 kW
    4. 47.4 kW
77. The work output from a reaction turbine stage is 280 kW per kg/s of steam. If the nozzle efficiency is 0.92 and rotor efficiency is 0.90, the isentropic static enthalpy drop will be
    1. 352 kW
    2. 347 kW

3. 338 kW
4. 332 kW

78. Shock waves in nozzles would occur while turbines are operating
    1. at overload conditions
    2. at part load conditions
    3. above critical pressure ratio
    4. at all off-design conditions
79. Consider the following statements:

    The erosion of steam turbine blades increases with the increase of

    1. moisture in the steam
    2. blade speed

    Which of these statement(s) is/are correct?

    1. 1 alone
    2. alone
    3. 1 and 2
    4. Neither 1 nor 2
80. Consider the following statements regarding an impulse turbine:

    1. Relative velocity at the inlet and exit of the rotor blades are the same.
    2. Absolute velocity at the inlet and exit of the rotor blades are the same.
    3. Static pressure within the rotor blade channel is constant
    4. Total pressure within the rotor blade channel is constant

    Of these statements,

    1. 1 and 4 are correct
    2. 2 and 3 are correct
    3. 1 and 3 are correct
    4. 2 and 4 are correct

81. If in an impulse turbine designed for free vortex flow, the tangential velocity of steam at the root radius of 250 mm is 430 m/s and the blade height is 100 mm, then the tangential velocity of steam at the tip will be
    1. 602 m/s
    2. 504 m/s
    3. 409 m/s
    4. 307 m/s
82. The blade passage for the nozzle blade row of the first stage of an impulse turbine is best represented an
    1.
    2.
    3.
    4.
    5.
83. If u, v_a, and u_r represent the peripheral, absolute, and relative velocities, respectively, and suffix 1 and 2 refer to inlet and outlet, which one of the following velocity triangles could be a reaction turbine stage with reaction more than 50%?
    1.
    2.
    3.
    4.

Explanatory Notes

1. 3. (b)

For superheated steam, n = 1.3

2. 4. (c)

Speed ratio =

Inlet velocity of steam, m/s

3. 7. (a)

Degree of reaction,

1. 10. (c) Let steam consumption, $y = A + Bx$

where x = kW

$$281 = A + 20\ B$$
$$521 = A + 50\ B$$

2. Solving we get, A = 121, B = 8

$$y = 121 + 15 \times 8 = 241 \text{ kg/h}$$

3. 19. (c) $u = 320$ m/s, $b_2 = 30°$
4. 23. (a) For maximum diagram efficiency,
5. 30. (d) Degree of reaction,
6. 35. (d) $P = \dot{m}\, u\, (v_{w1} - v_{w2})/10^3$ kW

Rate of change of momentum =

7. 36. (b)cm
8. 38. (c) Degree of reaction
9. 57. (b) Power developed in the last row kW

where n = number of rows

10. 62. (a) Work done by 1 kg of steam N.m
11. 63. (d) $u = v_w = v_{w1} + v_{w2} = 75$

Specific work $= u\,(v_{w1} + v_{w2}) = u^2 = 75^2 = 5625\ \text{m}^2/\text{s}^2$

12. 65. (a) Reheat factor
13. 71. (d) Tangential force $= \dot{m}\,(v_{w1} - v_{w2}) = 1(910 - 0) = 910$ N

Axial thrust $= \dot{m}(v_{f1} - v_{f2}) = 1 \times 4 = 4$N

14. 74. (a)
15. 76. (a) $v_{a1} = 236$ m/s, $v_{r1} = 132$ m/s, $v_{r2} = 232$ m/s, $v_{a2} = 136$ m/s

Specific work output

16. 77. (c) P = 280 kW, $h_n = 0.92$, $h_b = 0.90$
17. 81. (d) $r_1 = 250$ mm, $u_1 = 430$ m/s, $h = 100$ mm

In a free vortex flow,

or $u_1 r_1 = u_2 r_2$

Review Questions

1. Explain the principle of operation of a steam turbine.
2. How do you classify steam turbines?
3. Compare impulse and reaction turbines.
4. What do you mean by compounding of steam turbines? What are the various methods of compounding impulse steam turbines?
5. Define blade efficiency and stage efficiency.
6. What is the condition for maximum blade efficiency of an impulse turbine considering blade friction?
7. Define speed ratio and write the expression for a turbine with *n* blade rings.
8. What are the advantages of velocity compounding?
9. What are the limitations of velocity compounding?
10. What is the necessity of compounding impulse turbine?
11. Define degree of reaction.
12. What is the condition for blade angles for 50% degree of reaction?
13. Write the formula for maximum blade efficiency of a reaction turbine.
14. Define reheat factor and turbine internal efficiency.
15. List the various losses in a steam turbine.
16. What are the various methods for governing of a steam turbine?
17. Explain the working of a back-pressure turbine.
18. What is an extraction turbine?
19. Explain co-generation.
20. What is the role of Labyrinth packing?

Exercises

7.1 In a single stage impulse turbine, the blade angles are equal and the nozzle

angle is 20°. The velocity coefficient for the blade is 0.83. Find the maximum blade efficiency possible.

If the actual blade efficiency is 90% of maximum blade efficiency, find the possible ratio of blade speed to steam speed.

[**Ans.** 0.813, 0.3235]

7.2 The following data refer to a compound impulse turbine having two rows of moving blades and one row of fixed blades in between them.

Nozzle angle = 15°, Exit velocity of steam from the nozzle = 450 m/s

Moving blades tip discharge angles = 30°

Fixed blade discharge angle = 20°

Friction loss in each blade rows = 10% of the relative velocity

Calculate the blade velocity, blade efficiency, and specific steam consumption for the turbine.

[**Ans.** 98m/s, 78.8%, 45 kg/kWh]

7.3 In a stage of impulse-reaction turbine, steam enters with a speed of 250 m/s at an angle of 30° in the direction of blade motion. The mean blade speed is 150 m/s when the rotor is running at 3000 rpm. The blade height is 10 cm. The specific volume of steam at nozzle outlet and blade outlet are 3.5 m^3/kg and 4 m^3/kg, respectively. The turbine develops 250 kW. Assuming the

efficiency of nozzle and blades together considered as 90% and carry over coefficient as 0.8, find (a) the enthalpy drop in each stage, (b) degree of reaction, and (c) stage efficiency.

[**Ans.** 29.3 kJ/kg, 0.39, 71%]

7.4 At a stage of reaction turbine, the mean diameter of the rotor is 1.4 m. The speed ratio is 0.7. Determine the blade inlet angle if the blade outlet angle is 20°. The rotor speed is 3000 rpm. Find the diagram efficiency, percentage increase in it, and rotor speed if the rotor is designed to run at the best theoretical speed, the exit angle being 20°.

[**Ans.** 53.8°, 90.5%, 3.65%, 93.8%]

7.5 In a 50% reaction turbine stage running at 50 rps, the exit angles are 30°

and the inlet angles are 50°. The mean diameter is 1 m. The steam flow rate is 10^4 kg/min and the stage efficiency is 85%. Calculate (a) the power output of the stage, (b) the specific enthalpy drop in the stage, and (c) the percentage increase in the relative velocity of steam when it flows over the moving blades.

[**Ans.** 11.6 kW, 82 kJ/kg, 52.2%]

7.6 A single row impulse turbine develops 130 kW at a blade speed of 180 m/s using 2 kg/s of steam. The steam leaves the nozzle at 400 m/s. The friction coefficient for blades is 0.9 and steam leaves the blades axially. Determine (a) the nozzle angle and (b) the blade angles at entry and exit, assuming no shock.

[**Ans.** 25.47°, 43.5°, 35.8°]

7.7 A simple impulse turbine has one ring of moving blades running at 150 m/s. The absolute velocity of steam at exit from the stage is 80 m/s at an angle of 75° from the tangential direction. The blade speed coefficient for blades is 0.85 and the rate of steam flowing through the stage is 3 kg/s. If the blades are equiangular, determine (a) the blade angles, (b) the nozzle angle, (c) the absolute velocity of steam issuing from the nozzle, and (d) the axial thrust.

[**Ans.** 24.35°, 14.5°, 362.4 m/s]

7.8 In a single stage impulse turbine, the nozzle angle is 20° and blade angles are equal. The velocity coefficient for blades is 0.85. Calculate the maximum

blade efficiency possible. If the actual blade efficiency is 92% of the maximum blade efficiency, find the possible ratio of blade speed to steam speed.

[**Ans.** 81.6%, 0.336 or 0.603]

7.9 In a single stage steam turbine saturated steam at 10 bar is supplied through a convergent-divergent steam nozzle of 20° angle. The mean blade speed is 400 m/s. The steam pressure leaving the nozzle is 1 bar. Find (a) the best blade angles if they are equiangular and (b) the maximum power developed by the turbine if 5 nozzles are used of 0.6 cm^2 area at the throat. Assume nozzle efficiency 90% and blade friction coefficient 0.87.

[**Ans.** 35.46°, 115.8 kW]

7.10 The first stage of an impulse turbine is compounded for velocity and has two rows of moving blades and one ring of fixed blades. The nozzle angle is 15° and leaving angles of blades are respectively, first moving 30°, fixed 20°, second moving 30°. The velocity of steam leaving the nozzle is 540 m/s. The friction loss in each blade row is 10% of the relative velocity. Steam leaves the second row of moving blades axially. Calculate (a) the blade velocity, (b) the blade efficiency, and (c) the specific steam consumption.

[**Ans.** 117.3 m/s, 78.6%, 31.39 kg/kWh]

7.11 The first stage of a turbine is a two-row velocity compounded impulse wheel. The steam velocity at inlet is 600

m/s and the mean blade speed is 120 m/s. The nozzle angle is 16° and the exit angles for the first row of moving blades, the fixed blades, and the second row of moving blades are 18°, 21° and 35°, respectively. Calculate (a) the blade inlet angles for each row, (b) the driving force for each row of moving blades and axial thrust on the wheel for a mass flow rate of 1 kg/s, (c) the diagram efficiency and diagram power for the wheel per kg/s steam, and (d) the maximum possible diagram efficiency. Assume blade velocity coefficient as 0.9 for all blades.

[**Ans.** First row: Moving blade 20°, fixed blade 24.5°, Second row: moving blade 34.5°; 875 N, 294 N; 42 N; 140.28 kW, 77.93%, 92.4%]

7.12 The following data refer to a stage of a Parson's reaction turbine:

Speed of turbine = 1500 rpm

Mean diameter of rotor = 1 m

Stage efficiency = 80%

Blade outlet angle = 20°

Speed ratio = 0.7

Calculate the isentropic enthalpy drop in the stage.

[**Ans.** 13.08 kJ]

7.13 A stage of a Parson's reaction turbine delivers dry saturated steam at 2.7 bar from the fixed blades at 90 m/s. The mean blade height is 4 cm, and the moving blade exit angle is 20°. The axial velocity of steam is 3/4 of blade

velocity at the mean radius. Steam is supplied to the stage at the rate of 2.5 kg/s. The effect of blade tip thickness on the annulus area can be neglected. Calculate (a) the wheel speed, (b) the diagram power, (c) the diagram efficiency, and (d) the enthalpy drop in this stage.

[**Ans.** 1823 rpm, 13.14 kW, 78.7%, 5.25 kJ/kg]

7.14 In a stage of a reaction turbine, the mean rotor diameter is 1.5 m, speed ratio = 0.72, blade outlet angle = 20° and rotor speed = 3000 rpm. Calculate (a) the diagram efficiency and (b) the percentage increase in diagram efficiency and rotor speed if the rotor is designed to run at the best theoretical speed, the exit angle being 20°.

[**Ans.** 91%, 2.96%, 3914.7 rpm]

7.15 The nozzle angle of a single stage impulse turbine is 20°. The moving blade angles are equal and the velocity coefficient for the blade is 0.83. Find the maximum blade efficiency possible. If the actual blade efficiency is 90% of the maximum blade efficiency, find the possible ratio of blade speed to steam speed.

[ESE, 1979]

7.16 The total power developed by a 20-stage reaction turbine is 11200 kW. Steam is supplied at 15 bar abs. and 300°C. The pressure of steam leaving the turbine is 0.1 bar abs. The turbine has a stage efficiency of 75% and a reheat factor of 1.05. Determine the

mass flow rate of steam through the turbine. Assume that all stages develop equal power.

[ESE, 1980]

7.17 At a certain stage of the turbine in Exercise 7.16, the pressure of steam is 1 bar and it is dry saturated. If the blade exit angle is 20° and the ratio of the blade speed to steam speed is 0.7, find the mean, diameter of the rotor of this stage, and the rotor speed. Assume the blade height as the mean drum diameter and neglect the thickness of the blades. The velocity diagrams for the blades are symmetrical.

[ESE, 1980]

7.18 An impulse stage of a steam turbine is supplied with dry and

saturated steam at 15 bar. The stage has a single row of moving blades running at 3600 rpm. The mean diameter of the blade disc is 9.0 m. The nozzle angle is 15° and the axial component of the absolute velocity leaving the nozzles is 93.42 m/s. The height of nozzles at their exit is 100 mm. The nozzle efficiency is 0.9 and the blades velocity coefficient is 0.965. The exit angle of the moving blades is 2° greater than that at the inlet. Determine (a) the blade inlet and outlet angles, (b) the isentropic heat drop in the stage, (c) the stage efficiency, and (d) power developed by the stage.

[ESE, 1982]

7.19 A single row impulse turbine stage running at 3000 rpm has a blade to

steam speed ratio of 0.48. The nozzles are set at angle of 15° with respect to the place of the blade disc. The mean velocity of the blades is 144 m/s. The moving blades are symmetrical about their axis parallel to the direction of rotation. The nozzle efficiency is 0.92 and the blade velocity coefficient is 0.97. The height of the nozzles at their exit section is 100 mm. Steam leaving in the nozzles is saturated and dry and its pressure is 10 bar. Determine (a) the isentropic heat drop in the stage, (b) the energy lost in the nozzles and the moving blades due to friction, (c) energy lost due to finite velocity of steam leaving the stage, (d) mass flow rate, (e) power developed by the stage, and (f) the efficiency of the stage.

Assume that steam is accelerated from rest in the nozzles.

[ESE, 1983]

7.20 An impulse stage of steam turbine has a mean diameter of 1.2 m. The speed of rotor is 3000 rpm. The mass flow rate of steam is 20 kg/s. Steam is supplied to the stage at 15 bar and 300°C, where it expands to 10 bar. Determine the efficiency ant the power output of the stage if the nozzle efficiency is 0.9 and the blade velocity coefficient is 0.92. Assume acceleration from rest for the steam expanding in the nozzle. Assume nozzle angle to be 25°.

[ESE, 1986]

7.21 Steam enters a single-stage impulse turbine at 380 m/s and the blade speed is

170 m/s. The steam flow rate is 2.2 kg/s and turbine develops 112 kW. Assume the blade velocity coefficient to be 0.8. For an axial discharge of steam, find (a) nozzle angle, (b) blade angles, and (c) diagram efficiency.

[ESE, 1987]

7.22 Steam enters in a stage of impulse-reaction turbine with a speed of 280 m/s at an angle of 22° in the direction of blade motion. The mean diameter of the rotor, which rotates at 3200 rpm, is 1.0 m. The blade height is 10 cm. The specific volume of steam at nozzle outlet and blade outlet are 3.6 m^3/kg and 4.1 m^3/kg, respectively. The turbine develops 462 kW. Find (a) the weight of steam used per second, (b) blade angle,

(c) enthalpy drop in each stage, (d) degree of reaction, and (e) stage efficiency. Assume combined efficiency of nozzles and blades as 90% and carry over coefficient as 0.8.

[ESE, 1988]

7.23 The nozzles of a Lavel turbine deliver steam at the rate of 0.9 kg/s with a velocity of 730 m/s to a set of blades revolving at the rate of 30000 rpm The diameter of the wheel is 11.5 cm. The nozzles are inclined at an angle of 20° to the plane of wheel rotation. Calculate (a) the diagram efficiency, (b) power developed by the blades, (c) energy lost in the blades per second, and (d) the condition for maximum efficiency of the turbine. Assume the blade velocity

coefficient as 0.72 and outlet blade angle 25°C.

[ESE, 1989]

7.24 The nozzles of a two-row velocity-compounded stage have outlet angles of 22° and utilise an isentropic enthalpy drop of 200 kJ per kg of steam. All moving and guide blades are symmetrical, and the mean blade speed is 150 m/s. Assuming an isentropic efficiency for the nozzles of 90%, find graphically all the blade angles and calculate the specific power output produced by the stage. The velocity at inlet to the stage can be neglected.

[ESE, 1991]

7.25 One stage of an impulse turbine consists of a converging nozzles ring

and one ring of moving blades. The nozzles are inclined at 22° to the blade whose tip angles are both 35°. If the velocity of steam at exit from the nozzle is 660 m/s, find the blade speed so that the steam shall pass on without shock. Find the diagram efficiency neglecting losses if the blades are run at this speed.

[ESE, 1992]

7.26 A single steam turbine is supplied with steam at 5 bar, 200°C at the rate of 50 kg/min. It exhausts into a condenser at a pressure of 0.2 bar. The blade speed is 400 m/s. The nozzles are inclined at an angle of 20° to the plane of the wheel and the outlet blade angle is 30°. Neglecting friction losses, determine the blade efficiency, the stage efficiency and

the power developed by the turbine.

[ESE, 1994]

ANSWERS TO MULTIPLE-CHOICE QUESTIONS

1. a
2. b
3. b
4. c
5. b
6. c
7. a
8. a
9. c
10. c
11. a
12. d
13. d
14. c
15. c
16. b
17. b
18. d
19. c
20. a
21. a
22. d
23. a
24. c
25. c
26. d
27. b
28. b
29. b
30. d
31. a
32. d
33. a
34. a

35. d
36. b
37. d
38. c
39. c
40. a
41. b
42. d
43. d
44. a
45. c
46. c
47. c
48. c
49. c
50. c
51. b
52. a
53. b
54. d
55. d
56. b
57. b
58. a
59. d
60. c
61. a
62. a
63. d
64. c
65. a
66. b
67. c
68. c
69. b
70. b
71. d
72. d
73. a
74. a
75. c
76. a
77. c

78. a
79. c
80. a
81. d
82. c
83. c

Chapter 8

Steam Condensers

8.1 ❒ DEFINITION

A condenser is a closed vessel in which steam is condensed by abstracting the heat and where the pressure is maintained below atmospheric pressure.

8.2 ❒ FUNCTIONS OF A CONDENSER

There are two main functions of a condenser, which are explained below:

1. To reduce the back pressure considerably upon the prime mover, and thereby increasing the work output of steam during expansion.

   Consider the ideal $p-v$ diagram of a non-condensing steam engine, as shown in Fig. 8.1(a), in which steam is supplied at pressure p_1 and the exhaust takes place at atmospheric pressure p_2. The work output is given by the area

1–2–3–4–5–1. If we reduce the back pressure p_2 to p'_2, which is below the atmospheric pressure, keeping p_1 same, then the work output will be given by the area 1–2–3–4′–5′–1, as shown in Fig. 8.1(b). This work output is obviously greater, than the previous work output. The additional work output is 4–4′–5′–5–4.

2. The condensate condensed from the exhaust steam in the condenser is free from impurities and can be directly used as feed water for the boiler without any treatment. This recovery of feed water is an incidental advantage.

Figure 8.1 *Effect of back pressure on work output of a steam engine*

8.3 ❒ ELEMENTS OF STEAM CONDENSING PLANT

A steam power plant using a condenser is shown in Fig. 8.2. The various components of a steam condensing plant are described below.

1. **Condenser:** It is a closed vessel heat exchanger in which steam coming from the turbine is condensed by abstracting heat by cooling water where the pressure is maintained below atmospheric pressure.
2. **Condenser cooling water pump:** The cooled water from the cooling tower is circulated through the condenser by this pump. This pump is not required if water from the cooling tower is not re-circulated.
3. **Make up water pump:** This pump circulates the required quantity of water through the condenser.
4. **Condensate extraction pump:** The function of the condensate

pump is to extract the condensate from the condenser where the pressure is below atmospheric pressure and feed it to the hot well, where the pressure is at atmospheric level.

5. **Hot well:** It is a sump between the condenser and the boiler where the condensate coming from the condenser is collected.
6. **Air-extraction pump:** The function of the air-extraction pump is to extract the air which has leaked into the condenser by various paths and to maintain the required vacuum inside.
7. **Boiler feed pump:** Its function is to pump the condensate from hot well into the boiler by increasing the pressure of condensate above the boiler pressure.
8. **Cooling tower:** It is essential when large amount of cooling water required is not available from the river throughout the year. Its function is to cool the hot water coming out of the condenser instead of discharging it to waste. The hot water is cooled by evaporative cooling.

Figure 8.2 *Steam power plant using a condenser*

8.4 ❐ TYPES OF STEAM CONDENSERS

Steam condensers may be broadly classified into two categories.

1. **Jet condensers:** They are contact-type condensers in which the steam to be condensed mixes with the cooling water and the temperature of the condensate and the cooling water is the same when leaving the condenser. The condensate cannot be recovered for use as feed water to the boiler. Heat transfer is by direct conduction between steam and water.

   These condensers may be further classified as follows:

   1. Low-level, counter flow type condenser
   2. Low-level, parallel flow type condenser
   3. High-level jet condenser
   4. Ejector condenser

2. **Surface condensers:** There is no direct contact between steam to be condensed and the circulating cooling water in surface condensers. Heat is transferred through a wall interposed between steam and cooling water. The temperature of condensate may be higher than the cooling water. The condensate can be used as feed water to the boiler.

   These condensers may be further classified as follows:

   1. Cooling water in tubes and steam surrounding it
   2. Central flow or regenerative condenser
   3. Evaporative condenser

8.4.1 Jet Condensers

1. **Low level counter-flow jet condenser:** This type of condenser is shown in Fig. 8.3. The cold water is drawn up in the condenser shell from the cooling pond due to the vacuum head created in the shell. The level of the cooling pond should not be much below the delivery pipe to the shell as no pump is used. The shell is arranged with two or three water trays with perforations to break up water into small jets. The exhaust steam and any mixed air enters at lower portion of the shell and tries to ascend up through the falling sprays. Thus, steam gets condensed and air ascends up and cools down. The air is removed by a separate suction pump at the top. The mixture of condensate and cooling water descends down through a vertical pipe to the extraction pump and pumped to the hot well. From the hot well, the boiler feed pump delivers water to the boiler and surplus water overflows to the cooling pond. Since the boiler feed and the cool injection water mix, this type of condenser may be used only where cheap pure water is available. The non-condensable air and gas are continuously removed by air pump suction to maintain vacuum.
2. **Low level parallel flow Jet condenser:** It is similar to the counter flow jet condenser except that steam enters the condenser shell at the top and just below it the cooling water is delivered. It is less efficient than the counter flow type.
3. **High level jet condenser:** This is also called Barometric condenser and is shown in Fig. 8.4. The shell is placed at a

height greater than 10.363 m, the barometric height of water column. If a pipe of height more than 10.363 m is held vertical with one end immersed in water vessel open to atmosphere and the other end subjected to suction pressure, the atmospheric pressure will hold the column of water in the pipe equal to the suction pressure. This fact is made use of in this condenser by making the tail pipe more than 10.363 m in height and thus making it impossible for any vacuum in the condenser to cause the water to rise high in the tail pipe of the water leg and flood the engine.

Figure 8.3 *Low level counter flow jet condenser*

Figure 8.4 *High-level jet condenser*

Figure 8.5 *Ejector condenser*

The height of the shell much above the barometric height requires a separate pump for injecting cool water. The condensate and water will fall under gravity to the hot well and maintain a column of water in the water leg, depending on the vacuum in the condenser. A separate air pump to remove air is employed.

4. **Ejector condenser:** It is a low-level type of condenser (as shown in Fig. 8.5). This condenser works on the principle that by discharging a smooth jet of cold water under a head of about 6 m through a series of converging guide cones, the steam and the associated air are drawn in through the hollow truncated cones and led to the diverging cones. In the converging cones, the pressure energy is partly converted to kinetic energy. In the diverging cones, the kinetic energy is again partly converted to pressure energy so as to obtain pressure greater than the atmospheric to enable the condensate and water mixture to be discharged to the hot well exposed to atmospheric pressure.

A non-return valve is fitted on the exhaust steam inlet to condenser so that water from the hot well

does not rush back to the engine in case of cold injected water failure.

8.4.2 Surface Condensers

Water Tube Surface Condenser

In this type of surface condenser, water flows in the tubes and steam surrounds it. Thus, the condensate is directly available as an ideal boiler feed. A surface condenser produces more vacuum and any type of cooling water can be used. However, it is bulky and the initial capital cost is higher than jet condensers. But, this is justified by the saving in the running cost due to high efficiency.

This type of condensers may be further classified as follows:

1. Single pass, two pass condensers

A longitudinal section of a two pass, down flow condenser is shown in Fig. 8.6. It consists of a cast iron shell, cylindrical in shape, with the two ends covered by cover plates. A nest of brass tubes is fixed in the two tube plates at the end. They are fixed by brass ferrules so that they can be replaced easily.The space between the tube plates and the cover plates is known as water boxes. One of the water boxes has baffle partitioning the box into two sections—one upper half and the other lower half.

The cold water is sent through the lower half section tubes and comes out through the upper half section tubes. Exhaust steam enters the shell at the top

and flows down surrounding the cold water tubes. The condensate is removed at the bottom by an extraction pump.

The surface condensers may work on the wet vacuum system or the dry vacuum system. For the dry vacuum system, three pumps are required—one for the circulating cooling water, the other for removing the condensate, and the third, the dry air pump, for removing air. The cross-section of such a dry vacuum system is shown in Fig. 8.7. The air exit is shielded from the down flow of steam by means of a baffle.

The main requirements of a surface condenser are as follows:

1. Air should be removed at a cooler section and it should be as far as possible dry.

2. The pressure drop of steam in the condenser should be minimum.

Figure 8.6 *Water tube surface condenser*

Figure 8.7 *Down flow surface condenser*

Central Flow or Regenerative Condenser

In such a condenser as shown in Fig. 8.8, the steam passages are all around the periphery of the shell. The exhaust steam converges to the centre of the shell where there are no tubes. The air is removed by the air pump suction from the centre. Some of the exhaust steam also passes to the centre and gets condensed. The condensate is reheated to nearly the temperature of the exhaust steam. The pressure drop of steam is also reduced due to large flow passages.

Figure 8.8 *Central flow surface condenser*

Evaporative Condenser

In evaporative condenser, steam circulates in pipes surrounded by water spray, as shown in Fig. 8.9. It is used when there is scarcity of cooling water. The cooling water is allowed to evaporate under a small partial pressure. It consists of gilled piping bent to form many rows and is placed vertically. Steam passes through these pipes. Pumped water is sprayed from the top and descends down forming a thin film over the pipes. A natural or forced air draught causes the evaporation of the water film. Steam while circulating inside the pipes, latent heat of evaporation is removed from steam and thereby the steam gets condensed. This mode of heat transfer reduces the cooling water requirement of condenser

to 10% of the requirement for surface condensers. The water particles carried with air due to high velocity of air are removed with the help of eliminator. This type of condenser works better in dry weather.

Figure 8.9 *Evaporative condenser*

8.5 ❐ REQUIREMENTS OF MODERN SURFACE CONDENSERS

1. The steam should be evenly distributed over the entire cooling surface of the condenser with minimum pressure loss.
2. There should be no undercooling of the condensate.
3. Water should be passed through the tubes and steam must surround the tubes from outside.
4. There should be no air leakage into the condenser to maintain proper vacuum.
5. Air must be cooled to the maximum possible extent before extraction.
6. For better thermal efficiency, the rise in temperature of the cooling water should be limited to 10°C.

8.6 ❐ COMPARISON OF JET AND SURFACE CONDENSERS

8.6.1 Jet Condensers

Advantages

1. Mixing of steam and cooling water is more intimate.
2. Less quantity of circulating water for condensing exhaust steam is required.
3. Condensing plant is simple and less costly.
4. Spare parts required are less.
5. Condensate extraction pump is not needed in barometric jet and ejector condensers.

Disadvantages

1. Condensate is wasted unless cooling water is pure.
2. In the low-level jet condenser, if the condensate extraction pump fails, then the engine would get flooded unless safe guarded.
3. Barometric condenser cost is high due to long piping.
4. There is vacuum loss in barometric condenser due to leakage in long piping.
5. There are limitations of vacuum of up to 650 mm of Hg due to liberation of dissolved gases in cooling water.
6. The air extraction pump requires comparatively high power.

8.6.2 Surface Condensers

Advantages

1. Vacuum obtained is comparatively very high, resulting in high efficiency.
2. Condensate can be used directly as boiler feed.
3. There is no need of water treatment plant.
4. It is more suitable for high capacity plants.

Disadvantages

1. It requires more space.

2. Maintenance cost is more.

8.7 ❒ VACUUM MEASUREMENT

A vacuum gauge is used to measure the vacuum in the condenser. Figure 8.10 shows a vacuum gauge. It consists of a glass tube containing mercury to indicate the barometric pressure and another mercury tube connected to the condenser and vacuum gauge. The vacuum gauge has readings marked in cm of Hg. It represents the height at which a column of mercury, the upper surface of which is in communication with the cylinder, will stand when supported by barometric pressure. The vacuum is dependent upon the barometric pressure and the absolute pressure in the condenser.

Figure 8.10 *Vacuum gauge*

Absolute pressure in the condenser = barometric pressure – vacuum gauge pressure

Usually, the vacuum gauge readings are corrected to standard barometer reading of 76 cm of Hg.

Thus,

Corrected vacuum in cm of Hg = 76 – absolute pressure in cm of Hg

= 76 – (actual barometric height – vacuum)

Let H_b = actual barometric height in cm of Hg

H_g = actual vacuum gauge reading in cm of Hg

$H_b - H_g$ = absolute pressure in condenser in cm of Hg

p = absolute condenser pressure in bar

Now, 76 cm of Hg = 1.01325 bar

8.8 ❐ DALTON'S LAW OF PARTIAL PRESSURES

Dalton's law states that

1. The total pressure in a container having mixture of gas and vapour is the sum of the partial pressure of the vapour at the common temperature and the partial pressure of the gas.

   If the degree of saturation of air at any point inside the condenser is 100%, the partial pressure of vapour is the same as the saturation pressure.

   Let $t°C$ = temperature of mixture in a container.

   p_a = absolute pressure of any gas or air at $t°C$.

   p_s = saturation pressure of any vapour corresponding to $t°C$.

   p_t = total pressure in the container

2. Each constituent of the mixture in the container occupies the whole volume of the container and exerts its own partial pressure.

Let V = volume of a container, m^3

m_a = mass of air in the container, kg

m_g = mass of water vapour in the container, kg

m = total mass of mixture in the container, kg

v_g = specific volume of saturated vapour at t°C and p_a in m^3/kg

v_a = specific volume of air at t°C in m^3/kg

Example 8.1

A vacuum of 67 cm of Hg was obtained with the barometer reading of 75 cm of Hg. The condensate temperature is 20°C. Correct the vacuum to a standard barometer of

76 cm and hence determine the partial pressure of air and steam. Also find the mass of air present with 1 kg of steam.

Solution

Corrected vacuum = 76 – (actual barometric reading – actual vacuum)

$$= 76 - (75 - 67) = 68 \text{ cm of Hg}$$

Pressure of dry saturated steam at 20°C from steam tables is 0.023385 bar

and specific volume is $v = 57.79$ m^3/kg

Total absolute pressure in the

condenser = 75 – 67 = 8 cm of Hg

Partial pressure of steam, p_g = 1.754 cm of Hg

∴ Partial pressure of air, p_a = 8 – 1.754 = 6.246 cm of Hg

According to Dalton's law the volume of air present per kg of steam

= 57.79 m^3/kg at 20°C

Also for air $p_aV = m_aRT$

8.9 ❒ MASS OF COOLING WATER REQUIRED IN A CONDENSER

Let $\dot{m}_w$ = mass flow rate of circulating water, kg/h

$\dot{m}_s$ = mass flow rate of steam condensed, kg/h

t = temperature of wet exhaust steam, °C

t_c = temperature of condensate, °C

h_f = enthalpy of water at t°C, kJ/kg

h_{fg} = enthalpy of evaporation of steam at t°C, kJ/kg

x = dryness fraction of exhaust steam

t_{iw} = inlet temperature of cooling water, °C

t_{ow} = outlet temperature of cooling water, °C

h_c = enthalpy of condensate, kJ/kg = c_{pw} t_c

c_{pw} = specific heat of cooling water, 4.1868 kJ/kg.K

Now Heat given by steam = Heat absorbed by water

For jet condenser,

$$t_{ow} = t_c$$

For surface condenser,

$$m_s\,(h_f + xh_{fg} - h_c) = m_w\, c_{pw}\,(t_{ow} - t_{iw})$$

Example 8.2

A 200 kW steam engine has a steam consumption of 10 kg/kWh. The back pressure of the engine is equal

to the condenser pressure of 0.15 bar. The condensate temperature is 32°C. The cooling water temperature at inlet and outlet are 20°C and 30°C, respectively. Calculate the mass of cooling water required per hour if the exhaust steam is dry saturated.

Solution

At 0.15 bar, $h_f = 225.91$ kJ/kg, $h_{fg} = 2373.1$ kJ/kg, $t_c = 32°C$, $t_{ow} = 30°C$, $t_{iw} = 20°C$

Total steam used per hour = 200 × 10 = 2000 kg

Total circulating water used = 58.88

$\times 2000 = 117760$ kg/h

8.10 ❐ AIR REMOVAL FROM THE CONDENSER

8.10.1 Sources of Air Infiltration in Condenser

1. Air may pass into the condenser with the steam.
2. Air leakage may happen through the packings.
3. The cooling water contains dissolved gases which are released under vacuum.

The amount of air leakage should not be more than 5–15 kg/10000 kg of steam.

8.10.2 Effects of Air Infiltration in Condensers

1. The leaked air in the condenser increases the back pressure on the prime mover, and consequently, the thermal efficiency of the plant is lowered.
2. The infiltrated air lowers the partial pressure of steam which means a lower saturation temperature and increased latent heat. Hence, it will require greater amount of cooling water.
3. Because of poor thermal conductivity of air, the rate of heat transfer from the vapour is reduced which requires increased surface area of tubes for a given condenser duty.

8.11 ❐ AIR PUMP

The function of the air pump is to maintain a desired vacuum in the

condenser by removing air and condensed steam. Edward's air pump is commonly used for this purpose.

8.11.1 Edward's Air Pump

This pump is shown in Fig. 8.11. It consists of a plunger or piston D having a conical head E which slides inside a barrel or cylinder liner C having a cover B which has a number of delivery valves A. The passage G is connected to the condenser. The barrel has a number of parts, F pierced in the cylinder liner C. These parts communicate to the air pump suction pipe. H is the water weir where the discharged condensate collects and overflows to the hot well. This retention of sufficient water above the cylinder head acts as a water seal

against leakage of air. J is the relief valve.

On the downward stroke of the piston, partial vacuum is created above the piston because the head valves are closed and sealed by water. As soon as the ports are uncovered, the water vapour and air push into the space above the piston. The condensate collected at the piston is displaced by further motion of the piston with conical head and pushed rapidly through the ports to the top of the piston. On the upward stroke, the water, air, and water vapour above the piston are compressed to pressure above the atmospheric pressure and discharged outside through the head valves.

Capacity of the pump,

where D = piston diameter, L = stroke length, N = rpm, η_v = volumetric efficiency.

Figure 8.11 *Edward's air pump*

8.12 ❐ VACUUM EFFICIENCY

The lowest pressure which can exist in a condenser is the saturation pressure of steam corresponding to the temperature of water entering the condenser. However, the actual pressure is always greater than the ideal pressure by an amount equal to the partial pressure of air present in the condenser.

Let p_s = saturation pressure of steam corresponding to the temperature of

water entering the condenser.

p_a = partial pressure of air in the condenser.

p_b = barometric pressure

$p_t = p_a + p_s$

= total pressure of air and steam in the condenser

Then ideal vacuum possible without air leakage = $p_b - p_s$

Actual vacuum existing in condenser due to air leakage

$= p_b - p_t = p_b - (p_a + p_s)$

The “vacuum efficiency” is defined as

the ratio of actual vacuum to ideal vacuum.

The factors affecting the vacuum efficiency are as follows:

1. Air leakage increases p_a, and hence, it decreases vacuum efficiency.
2. The vacuum efficiency decreases with increase in barometric pressure, keeping p_a and p_s the same.
3. With insufficient cooling water, the pressure in the condenser increases and subsequently, reduces the vacuum efficiency of the condenser.

8.13 ❐ CONDENSER EFFICIENCY

It is defined as the ratio of the difference between the outlet and inlet temperatures of the cooling water to the difference between the temperature t_s corresponding to the vacuum in the condenser and the inlet temperature of cooling water.

where t_{iw}, t_{ow} = temperature at inlet and outlet respectively of cooling water, t_s = temperature of steam corresponding to the actual absolute pressure in the condenser.

8.14 ❐ COOLING TOWER

There is a need to recycle the cooling water through the condenser when cooling water supply is limited. Cooling tower is an artificial device used to cool hot cooling water coming out from the condenser.

There are different types of cooling towers. The induced draft type cooling towers are commonly used in high capacity power plants. The schematic diagram of an induced draft cooling tower is shown in Fig. 8.12. The hot

water coming out from the condenser is sprayed at the top of the tower and air is induced to flow through the tower with the help of induced draft fans mounted at the top of the tower.

Figure 8.12 *duced draft cooling tower*

The amount of water supply lost in such a tower ranges from 1% – 2% by evaporation and 0.5% – 2% by drift losses. To compensate these losses, make-up water is supplied from external sources. The cooled water is collected at the bottom.

Example 8.3

The following readings were taken during a test on a surface condenser:

Vacuum in condenser = 71 cm of Hg

Barometer reading = 76 cm of Hg

Temperature in condenser = 33°C

Hot well temperature = 30°C

Cooling water circulated = 48,000 kg/h

Inlet temperature of cooling water = 15°C

Outlet temperature of cooling water = 28°C

Condensate = 1500 kg/h

Calculate:

1. the mass of air in kg/m^3 of condensate volume,
2. dryness fraction of steam entering the condenser, and
3. the vacuum efficiency

Solution

1. Absolute pressure in condenser,

   $p_t = (76 - 71) \times 0.01333 = 0.0667$ bar

   At $t = 33°C$, $p_s = 0.05106$ bar from steam tables = 3.83 cm of Hg

   Partial pressure of air, $p_a = p_t - p_s = 0.0667 - 0.05106 = 0.01564$ bar

2. At $p = 0.0667$ bar, $h_f = 159.47$ kJ/kg, $h_{fg} = 2411.4$ kJ/kg

3. Vacuum efficiency = 0.9837 or 98.37%

Example 8.4

A steam turbine discharges 6000 kg of steam per hour at 45°C and 0.82 dryness. The estimated air leakage is 15 kg/h. The temperature at the

suction of air pump is 30°C and temperature of condensate is 33°C. Find (a) the vacuum gauge reading, (b) capacity of air pump, (c) loss of condensate per hour, and (d) the quantity of cooling water required by limiting its temperature rise by 10°C.

Solution

1. Total pressure in the condenser, $p_t = p_a + p_s$

At 45°C, $p_s = 0.095934$ bar from steam tables

$v_g = 15.258\ m^3/kg$

Volume of 6000 kg steam, $V = 6000 \times x v_g = 6000 \times 0.82 \times 15.258 = 75069.36\ m^3/h$

$= \text{volume of air}$

Partial pressure of air,

$$p_t = 18.236 \times 10^{-5} + 0.09582 = 0.096 \text{ bar}$$

Vacuum in condenser

2. Partial pressure of steam at air pump suction at 30°C = 0.04242 bar

$p_a = p_t - p_s = 0.096 - 0.042461 = 0.0535$ bar

Volume of air at 30°C and 0.0535 bar,

Air pump capacity = 243.82 m^3/h

3. Specific volume of steam at 30°C, $v_g = 32.893$ m^3/kg
4. At 45°C, $h_f = 188.42$ kJ/kg, $h_{fg} = 2394.8$ kJ/kg

Mass of cooling water,

= 288.62 tonnes/h

Example 8.5

The air leakage into a surface condenser operating with a steam turbine is estimated as 80 kg/h. The vacuum near the inlet of air pump is 70 cm of Hg when barometer reads 76 cm of Hg. The temperature at the inlet of vacuum pump is 20°C. Calculate (a) the minimum capacity

of the air pump, (b) dimensions of the reciprocating air pump to remove air if it runs at 200 rpm. Take L:D:: 3:2 and volume efficiency = 98%, (c) mass of vapours extracted per hour.

Solution

1. Total pressure at inlet of air pump at 1–1, (Fig. 8.13)

   Partial pressure of water vapour at 1–1 corresponding to saturation temperature of 20°C, from steam tables,

   $p_s = 0.023385$ bar

   Partial pressure of air at 1–1,

   $p_a = p_t - p_s = 0.08 - 0.023385 = 0.0566$ bar

   Volume of 80 kg/h of air at 20°C and 0.0566 bar,

   Capacity of air pump = 1188.6 m^3/h

2. Now $V = D^2 L N \eta_v$

$D^3 = 0.0858$

$D = 0.441$ m or 46.1 cm

$L = 66.2$ cm

Figure 8.13 *Surface condenser*

3. Mass of water vapour going with air in the air pump,

Example 8.6

The quality of steam entering a jet condenser is 0.92. It is condensed by using cooling water at 15°C. The mass of air in the condenser is 35% of the mixture. Assuming that only latent heat of steam is absorbed by the cooling water, calculate (a) the temperature of mixture of condensate and water leaving the condenser, and (b) the mass of water required per kg of steam

condensed.

Take for air, R = 287 J/kg.K and for steam, R = 463 J/kg.K. Condenser vacuum = 61 cm of Hg, Barometer reading = 76 cm of Hg.

Solution

1. Given that $p_a = 0.35\ p_t$

   $p_t = (76 - 61) \times 0.0133 = 0.2$ bar

   $p_a = 0.35 \times 0.2 = 0.07$ bar

   $p_s = p_t - p_a = 0.2 - 0.07 = 0.13$ bar

   Saturation temperature of steam at 0.13 bar, from steam tables,

   $$t_s = 50.71°\text{C}$$

   As only latent heat of steam is to be absorbed, therefore, outlet temperature of cooling water,

   $$t_{ow} = t_s = 50.71°\text{C}$$

   Since steam and cooling water mix together in the jet condenser,

Mixture temperature coming out of condenser = 50.71°C

2. Now

At p_s = 0.13 bar, h_f = 212.27 kJ/kg, h_{fg} = 2380.98 kJ/kg from steam tables

Minimum quantity of cooling water required per kg of steam condensed

= 14.65 kg

Example 8.7

In a trial on a surface condenser, the vacuum gauge reading was 660 mm of Hg when barometer read 760 mm of Hg. The temperature in the condenser was 45°C.

1. Determine the alteration in vacuum if the quantity of air entering the condenser is reduced by 100 kg/h.
2. If the mass of condensate is 960 kg/h, calculate the quantities of air and vapour which the pump has to handle.

Solution

Absolute condenser pressure = 760 – 660 = 100 mm of Hg

$$p_t = \times 1.01325 = 0.1333 \text{ bar}$$

At 45°C, from steam tables,

Partial pressure of steam, p_s = 0.095934 bar

Specific volume of steam, v_{sg} = 15.258 m^3/kg

Partial pressure of air, p_a = 0.1333 – 0.095934 = 0.037366 bar

Volume of steam per hour, V_s = 960 × 15.258 = 14647.68 m^3

$$= \text{Volume of air, } V_a$$

If the mass of air is reduced to 100 kg/h, then partial pressure of air,

New absolute pressure of condenser, $p'_t = 0.095934 + 0.00623 = 0.102$ bar

New vacuum reading = 760 – 76.52 = 683.48 mm Hg

Alteration in vacuum = 683.48 – 660 = 23.48 mm Hg

Capacity of pump = 960 + 100 = 1060 kg/h

Example 8.8

The following observations were recorded during a condenser test;

Vacuum reading = 700 mm of Hg; Barometer reading = 760 mm of Hg;

Condensate temperature = 34°C

Find: (a) partial pressure of air and (b) mass of air per m^3 of condenser volume.

Take R for air = 287 J/kg K

Solution

Given that Vacuum reading = 700 mm of Hg: Barometer reading = 760 mm of Hg; T = 34°C = 34 +

273 = 307 K

1. Pressure in the condenser

p_c = Barometer reading – Vacuum reading

= 760 – 700 = 60 mm of Hg

= 60 × 0.00133 = 0.0798 bar (∵ 1 mm of Hg = 0.001 33 bar)

From steam tables, corresponding to a temperature of 34°C, we find that pressure of steam,

$$p_s = 0.0535 \text{ bar}$$

∴ Partial pressure of air,

$$p_a = p_c - p_s = 0.0798 - 0.0535 = 0.0263 \text{ bar}$$

2. Mass of air per m^3 of condenser volume.

Example 8.9

A vacuum gauge fitted to a condenser reads 680 mm of Hg,

when the barometer reads 750 mm of Hg. Determine the corrected vacuum in terms of mm of Hg and bar.

Solution

Given that Vacuum gauge reading = 680 mm of Hg; Barometer reading = 750 mm of Hg:

Pressure in the condenser = 750 – 680 = 70 mm of Hg

Corrected vacuum = 760 – 70 = 690 mm of Hg

$$= 690 \times 0.00133 = 0.918 \text{ bar}$$

Example 8.10

Calculate the vacuum efficiency from the following data:

Vacuum at steam inlet to condenser = 700 mm of Hg

Barometer reading = 760 mm of Hg
Hot well temperature = 30°C

Solution

Given that Vacuum reading or actual vacuum = 700 mm of Hg; Barometer reading = 760 mm of Hg; t = 30°C

Pressure in the condenser = 760 – 700 = 60 mm of Hg

From steam tables, corresponding to a temperature of 30°C, we find that ideal pressure of steam,

Ideal vacuum = Barometer reading – Ideal pressure

= 760 – 31.93 = 728.07 mm of Hg

Vacuum efficiency, η_{vac} = = 0.9614 or 96.14%

Example 8.11

The vacuum efficiency of a condenser is 96%. The temperature of condensate is 40°C. If the barometer reads 752 mm of Hg, find the vacuum gauge reading of the condenser.

Solution

Given that η_v = 96% = 0.96%; t = 40°C;

We find that ideal pressure of steam, = 0.073837 bar = = 55.52 mm of Hg

∴ Ideal vacuum = Barometer reading – Ideal pressure

$$= 752 - 55.52 = 696.48 \text{ mm of Hg}$$

∴ Actual vacuum or vacuum gauge reading of the condenser

$$= 0.96 \times 696.48 = 668.62 \text{ mm of Hg}$$

Example 8.12

In a surface condenser, the vacuum

maintained is 700 mm of Hg. The barometer reads 754 mm. If the temperature of condensate is 18°C, determine: (a) mass of air per kg of steam and (b) vacuum efficiency.

Solution

Given that Actual vacuum = 700 mm of Hg; Barometer reading = 754 mm of Hg; T = 18°C = 18 + 273 = 291 K

Pressure in the condenser, p_c = 754 – 700 = 54 mm of Hg

From steam tables, corresponding to 18°C, we find that absolute or ideal pressure of steam,

Specific volume of steam, v_s = 65.844 m^3/kg

1. Pressure of air (as per Dalton's law)

   $p_a = p_c - p_s = 54 - 15.68 = 38.32$ mm of Hg

   $= 38.32 \times 0.00133 = 0.051$ bar $= 0.051 \times 10^5$ N/m^2

   Mass of air per kg of steam,

2. Ideal vacuum = Barometer reading – Ideal pressure
   = 754 – 15.68 = 738.32 mm of Hg

Example 8.13

A surface condenser, fitted with separate air and water extraction pumps, has a portion of the tubes near the air pump suction screened off from the steam so that the air is cooled below the condensate temperature. The steam enters the condenser at 38°C and the

condensate is removed at 37°C. The air removed has a temperature of 36°C. (a) If the total air infiltration from all sources together is 5 kg/h, determine the volume of air handled by the air pump per hour. (b) What would be the corresponding value of the air handled if a combined air and condensate pump was employed? Assume uniform pressure in the condenser.

Solution

Given that $T_s = 38°C = 38 + 273 = 311$ K; $T_c = 37°C = 37 + 273 = 310$ K; $T_a = 36°C = 36 + 273 = 309$ K; $\dot{m}_a = 5$ kg/h

1. Since the pressure at entry to the condenser (p_c) is equal to the pressure of steam corresponding to 38°C, therefore, from

the steam tables,

$$p_c = 0.0668 \text{ bar}$$

Pressure of steam at the air pump suction, corresponding to 36°C (from steam tables),

$$p_s = 0.0598 \text{ bar}$$

∴ Pressure of air at the air pump suction (as per Dalton's law),

$$p_a = p_c - p_s = 0.0668 - 0.0598 = 0.007 \text{ bar}$$
$$= 0.007 \times 105 = 700 \text{ N/m2}$$

Volume of air handled by the air pump,

2. From steam tables corresponding to a condensate temperature of 37°C, we find that pressure of steam,

$$p_s = 0.0633 \text{ bar}$$

∴ Pressure of air (as per Dalton's law),

$$p_a = p_c - p_s = 0.0668 - 0.0633 = 0.0035 \text{ bar}$$
$$= 0.0035 \times 105 = 350 \text{ N/m2}$$

Volume of air handled,

Example 8.14

The air leakage into a surface condenser operating with a steam

turbine is estimated as 84 kg/h. The vacuum near the inlet of air pump is 700 mm of Hg when barometer reads 760 mm of Hg. The temperature at inlet of vacuum pump is 20°C. Calculate the following:

1. The minimum capacity of the air pump in m^3/h;
2. The dimensions of reciprocating air pump to remove the air if it runs at 200 rpm. Take L/D ratio = 1.5 and volumetric efficiency = 100%
3. The mass of vapour extracted per minute

Solution

Given that $\dot{m}_a$ = 84 kg/h; Vacuum = 700 mm of Hg; Barometer reading = 760 mm of Hg; T = 20°C = 20 + 273 = 293 K

1. Pressure in the condenser,

p_c = Barometer reading – Condenser vacuum

$= 760 - 700 = 60$ mm of Hg

$= 60 \times 0.00133 = 0.0798$ bar

From steam tables, corresponding to a temperature of 20°C, we find that pressure of steam,

$$p_s = 0.023385 \text{ bar}$$

∴ Pressure of air (as per Dalton's law),

$$p_a = p_c - p_s = 0.0798 - 0.023385 = 0.056415 \text{ bar}$$
$$= 0.056415 \times 105 = 5641.5 \text{ N/m2}$$

Minimum capacity of the air pump,

2. Let D = Diameter of the cylinder in metre,

L = Length of the stroke in metres = 1.5 D

η_v = Volumetric efficiency = 100% = 1

N = Speed of the pump = 200 r.p.m.

Minimum capacity of the air pump (V_a),

∵ $D^3 = 0.0886$ or $D = 0.446$ m

and $L = 1.5\, D = 1.5 \times 0.446 = 0.669$ m

3. From steam tables, corresponding to a temperature of 20°C, we find that specific volume of steam,

$$v_g = 57.79 \text{ m3/kg}$$

∴ Mass of vapour extracted per minute

Example 8.15

The vacuum at the extraction pipe in a condenser is 710 mm of mercury and the temperature is 35°C. The barometer reads 760 mm of mercury. The air leakage into the condenser is 4 kg per 10,000 kg of steam. Determine: (a) the volume of air to be dealt with by the dry air pump per kg of steam entering the condenser, and (b) the mass of water vapour associated with this air. Take R = 287 J/kg K for air.

Solution

Given that vacuum = 710 mm of Hg; T = 35°C = 35 + 273 = 308 K; Barometer reading = 760 mm of Hg; m_a = 4 kg per 10000 kg of steam = 0.0004 kg/kg of steam.

1. Let V_a = Volume of air per kg of steam entering the condenser.

   Pressure in the condenser

   $$p_c = \text{Barometer reading} - \text{Condenser vacuum}$$
   $$= 760 - 710 = 50 \text{ mm of Hg} = 50 \times 0.001\,33$$
   $$= 0.0065 \text{ bar}$$

   From steam tables, corresponding to the temperature of 35°C, we find that the pressure of steam,

   $$p_s = 0.05628 \text{ bar}$$

   ∴ Pressure of air (as per Dalton's law),

   $$p_a = p_c - p_s = 0.0665 - 0.05628 = 0.01022 \text{ bar}$$
   $$= 0.01022 \times 105 = 1022 \text{ N/m2}$$

   Now $p_a V_a = m_a RT$

2. From steam tables, corresponding to a temperature of 35°C, we find that specific volume of steam,

   $$v_g = 25.216 \text{ m3/kg}$$

   ∴ Mass of water vapour associated with the air

Example 8.16

The amount of steam condensed in a condenser is 6800 kg/h and the air leakage into the condenser is 12 kg/h. The air pump suction is screened off. The exhaust steam temperature is 32°C, the condensate temperature is 30°C and the temperature at the air pump suction is 25°C. Determine the following:

1. The mass condensed in the air cooler section per hour
2. The volume of air in m^3 handled by air pump
3. The percentage reduction in air pump capacity due to cooling of the air.

Solution

The saturation pressure corresponding to steam temperature

32°C from steam table is

$p_a = 0.04799$ bar

Specific volume corresponding to 32°C, $v_s = 29.82\ m^3/kg$

∴ Volume of steam handled, $V_s = 6800 \times 29.82 = 202776\ m^3/h$

According to Dalton's law, the volume of air handled per hour is equal to the volume of steam handled per hour. Hence the partial pressure of air,

We observe that the value of partial pressure of air is very small and hence the total condenser pressure may be taken as due to steam alone

i.e. 0.04799 bar.

Since the temperature at the air pump suction is 25°C, hence the steam pressure at the air pump suction corresponding to 25°C from steam tables,

$$p_s = 0.031691 \text{ bar}$$

Specific volume corresponding to 25°C, $v_s = 43.359 \text{ m}^3/\text{kg}$

∴ Partial pressure of air handled,

$$p_a = 0.04799 - 0.031691 = 0.0163 \text{ bar}$$

∴ Volume of air handled per hour by the pump,

According to Dalton's law, the volume of steam will also be equal

to 629.64 m^3/h.

The temperature may be assumed to be that of the condensate, i.e., t_s = 30°C at the point where the saturated steam enters the air cooling section. Corresponding to 30°C, the pressure is:

$$p_s = 0.042461 \text{ bar}$$

Specific volume, v_s = 32.893 m^3/h

∴ Partial pressure of air is given by,

$$p_a = (0.04799 - 0.042461) = 0.00553 \text{ bar}$$

Hence, the volume of air handled

According to Dalton's law, this volume of air is equal to the volume

of steam hence the mass of steam associated with the air,

∴ Mass of steam condensed in the cooler section

$$= 57.37 - 14.52 = 42.85 \text{ kg/h}$$

Hence, the percentage reduction in volume handled by the air pump due to cooling section is:

Example 8.17

In a condenser test, the following observations were made

Vacuum = 70 cm of Hg

Barometer = 76 cm of Hg

Mean temperature of condensation = 35°C

Hot well temperature = 29°C

Mass of cooling water = 45500 kg/h

Inlet water temperature = 16.5°C

Outlet water temperature = 31°C

Mass of condensate = 1200 kg/h

Find (a) the mass of air present per unit condenser volume, (b) the state of steam entering the condenser, (c) the vacuum efficiency, (d) the condensate undercooling and (e)

condenser efficiency.

Solution

1. Corresponding to 35°C, the partial pressure of steam from steam table = 0.05628 bar

   Absolute pressure of mixture in condenser = (76 − 70) × = 0.07997 bar

   ∴ Partial pressure of air = 0.07997 − 0.05628 = 0.02369 bar

   Mass of air =

2. Heat absorbed by circulating water = 45500 × 4.18 (31 − 16.5) = 2727755 kJ/h

   Heat rejected by steam in condensing at 35°C and in being under cooled from 35°C to 29°C

   $$= m_c\ (h_{w1} + xL_1 - h_{wc})$$

   $$= 1200\ (146.66 + 2418.6x - 121.59)$$

   $$= 1200\ (25.07 + 2418.6x)$$

   Now, Heat absorbed by circulating water = Heat rejected by steam

3. Vacuum efficiency
4. Condensate undercooling = 35 − 29 = 6°C

5. Condenser efficiency =

Example 8.18

In a surface condenser 2500 kg/h of steam is condensed and the air leakage is 2 kg/h. A separate air and water extraction pumps are fitted which draws the air in air cooler and a portion of the tube near the air pump suction is screened off from the steam so that the air is cooled below the condensate temperature. The steam enters the condenser dry and saturated at 38°C and the condensate is extracted at the lowest point of the condenser at a temperature of 37°C (i.e., the temperature at the entrance of air cooler is 3°C). The temperature at

the air pump suction is 31°C. Assuming a constant vacuum throughout the condenser, find the following:

1. The mass of steam condensed/min in the air cooler
2. The volume of air to be dealt with per minute by the dry pump
3. The reduction in the necessary air pump capacity following the cooling of air.

Solution

Partial pressure of steam at 38°C, $p_s = 0.0668$ bar

Specific volume of steam at 38°C, $v_s = 21.8$ m^3/kg

This must be the volume of 2 kg/h of air when exerting its partial pressure, i.e., $V_a = V_s = 908.33$ m^3/min

Partial pressure of air, p_a = = 0.000033 bar

Hence, total pressure in the condenser = $p_s + p_a$ = 0.0668 + 0.000033 = 0.06683 bar

At condensate extraction:

Partial pressure of steam at 37°C = 0.0633 bar

Specific volume of steam at 37°C, v_s = 22.94 m^3/kg

∴ Partial pressure of air = 0.06683 – 0.0633 bar = 0.00353 bar

∴ Volume of air, $\dot{V}_a$ = = 8.473 m^3/min = V_s

∴ The mass of the steam associated with air,

At air pump suction:

Partial pressure of steam of 31°C = 0.04522 bar

∴ Partial pressure of air = 0.06683 - 0.04522 = 0.02161 bar

1. Mass of steam associated with this air and which is partially condensed in the air cooler

2. Mass of steam condensed in the air cooler or saving in condensate by using the separate extraction method

$$= 0.369 - 0.0429 = 0.3261 \text{ kg/min}$$

3. Air capacity without air cooler = 8.473 m^3/min

Air pump capacity with air cooler = 1.346 m^3/min

∴ Percentage reduction in pump capacity

Example 8.19

A barometric jet condenser maintains a vacuum of 64.2 cm of Hg when the barometer reads 76 cm of Hg. The condenser handles 4600 kg of steam/h and 0.97 dryness fraction. The inlet temperature of the cooling water is 15°C. The mixture of condensate and cooling water leaves at 43°C. Calculate (a) the minimum height of the tail pipe above the level of the hot well and (b) the amount of cooling water required. Assume no undercooling.

Solution

1. Absolute condenser pressure of steam = (76 − 64.2) × 0.0133 = 1.6048 m of water

   Barometer height = 76 × = 10.336 m of water

Hence the length of the tail pipe = 10.336 – 1.6048 = 8.7312 m

2. The saturation temperature of steam entering the condenser will be 43°C in the absence of any undercooling of the condensate. Corresponding to 43°C,

Example 8.20

A condenser is equipped in a steam turbine which handles 14500 kg of steam per hour and develops 2484.3 kW. The initial conditions of steam entering to turbine are 27 bar and 350°C. The exhaust from the turbine is condensed in the condenser and the vacuum maintained is 72.5 cm of Hg while barometer reads 75.8 cm of Hg. The temperature of the circulating water is increased from 8 to 28°C while the condensate is removed from the

condenser at a temperature of 29°C. Calculate the following:

1. Dryness fraction of steam entering the condenser
2. Mass of circulating water per hour and the cooling ratio
3. Minimum quantity of cooling water required per kg of steam.

Solution

1. Absolute condenser pressure = (75.8 − 72.5) × 1.013/76 = 0.04398 bar

   From the steam tables, corresponding to 0.043398 bar, h_f = 327.19 kJ/kg and h_{fg} = 2313.46 kJ/kg

   Corresponding to 27 bar and 350°C, h = 3124.84 kJ/kg

   If x be the dryness fraction of steam entering the condenser, then drop of enthalpy

   $\therefore x = 0.9414$

2. Heat given by steam = Heat received by water

   $14500[(121.48 + 0.9414 \times 2433.1) - 117.31] = m_w \times 4.18\ (28 - 8)$

   Mass of cooling water = m_w = 398002.5 kg/h

3. The maximum possible temperature of cooling water at outlet will be 29.4°C corresponding to condenser pressure of 0.04398 bar.

   Hence maximum rise in temperature of cooling water = 29.4 – 8 = 21.4°C

   If under this condition m_{w1} be the quantity of cooling water required per kg of steam, by the heat balance equation, then

   $[(123.22 + 0.942 \times 2432.4 - 121.48)]\ m_{w1} \times 4.18 \times 21.4$

$$m_{w1} = 25.634 \text{ kg of water/kg of steam}$$

Example 8.21

The following observations were recorded during a trial on a steam condenser:

Condenser vacuum recorded = 70 cm (0.93325 bar)

Barometer reading = 76.5 cm Hg

(1.02 bar)

Mean condenser temperature = 35°C

Hot well temperature = 28°C

Condensate formed/hr. = 1800 kg

Circulating cooling water inlet temperature = 15°C

Circulating cooling water outlet temperature = 27°C

Quantity of cooling water/h = 80000 kg

Determine the following:

1. Vacuum corrected to standard barometer of 76 cm Hg (1.01325 bar)

2. Vacuum efficiency
3. Under cooling of condensate
4. Condenser efficiency
5. State of steam entering condenser
6. Mass of air per cubic metre of condenser volume.
7. Mass of air present per kg of uncondensed steam

Assume *R* for air as 287 kJ/kgK.

Solution

1. Vacuum corrected to standard barometer
   = Standard barometric pressure – (Barometric pressure – Gauge pressure)
   = 1.01325 bar – (1.02 bar – 0.93335 bar)
   = 1.01325 – 0.8665 = 0.9266 bar
2. From steam table, the pressure corresponding to condenser temperature of 35°C = 0.0562 bar
3. Undercooling of condensate = 35° – 28° = 7°C
4. Absolute condenser pressure = Barometer pressure – Vacuum reading

   = 1.02 – 0.93325 = 0.08675 bar

   Saturation temperature corresponding to pressure of 0.08665 bar from steam tables is 42.9°C

   Thus, cooling water temperature can get raised to 42.9°C

5. From steam tables for pressure of 0.08665 bar
   h_f = 179.8 kJ/kg, hfg = 2399.6 kJ/kg.

   Enthalpy of condensate corresponding to hot

well temperature of 28°C is 117 kJ/kg.

Heat absorbed by cooling water = Heat given by steam

$$mc_{pw}\,(t_2 - t_1) = m_s\,(h_f + xh_{fg} - h_c)$$

6. At condenser temperature of 35°C, the partial pressure of steam,
p_s = 0.0562 bar.

Hence, m_a = = 1.911 kg per kg of vapour

or m_v = = 0.523 kg of vapour per kg of air.

With external cooling from 25°C to 20°C we have, from steam tables, at 20°C

p_s = 0.023366 bar = 1.7526 cm Hg, v_g = 57.834 m3

$\therefore p_a = 5.2 - 1.7526 = 3.4474$ cm Hg

7. where p_a = 3.4474 × = 0.04596 bar

$V_a = v_g$ = 57.834 m^3 per kg of vapour

R = 287 J/kg ×K; T = (273 + 20) = 293 K

Hence, m_a = = 3.161 kg per kg of vapour

m_v = = 0.316 kg of vapour per kg of air

These calculations show that by cooling the mixture form 25°C to 20°C, vapour accompanying each kg of air withdrawn from the condenser is reduced from 0.523 kg to 0.316 kg. This will have a significant effect both on the ejector capacity and work done for air pump suction.

Example 8.22

In a condenser test, the following observations were made:

Vacuum = 690 mm of Hg; Barometer reading = 750 mm of Hg; Mean temperature of condensation = 35°C; Hot well

temperature = 28°C; Mass of cooling water = 50,000 kg/h; Inlet temperature = 17°C; Outlet temperature = 30°C; Mass of condensate per hour = 1250 kg.

Find: (a) the mass of air present per m^3 of condenser volume, (b) the state of steam entering the condenser and (c) the vacuum efficiency.

Take R for air = 287 J/kg K.

Solution

Given that vacuum = 690 mm of Hg; Barometer reading = 750 mm of Hg; t_c = 35°C; t_h = 28°C; $\dot{m}_w$ = 50000 kg/h; t_f = 17°C; t_o = 30°C;

$\dot{m}_s = 1250$ kg/h; R = 287 J/kg K

1. We know that pressure in the condenser,

$$p_c = 750 - 690 = 60 \text{ mm of Hg} = 60 \times 0.00133 = 0.08 \text{ bar}$$

From steam tables, corresponding to a condensation temperature of 35°C, we find that the pressure of steam,

$$p_s = 0.05628 \text{ bar}$$

∴ Pressure of air (as per Dalton's law),

$$p_a = p_c - p_s = 0.08 - 0.05628 = 0.02372 \text{ bar} = 0.02372 \times 105 = 2372 \text{ N/m2}$$

Mass of air per m^3 of condenser volume,

2. Let x = Dryness fraction (i.e., state) of steam entering the condenser.

From steam tables, corresponding to a pressure of 0.05628 bar (or 35°C), we find that,

$$h_f = 146.66 \text{ kJ/kg and hfg} = 2418.6 \text{ kJ/kg}$$

Corresponding to a hot well temperature of 28°C,

$$h_{f1} = 117.3 \text{ kJ/kg}$$

Total heat of entering steam,

$$h = h_f + xh_{fg} = 146.66 + \text{x} \times 2418.6 \text{ kJ/kg}$$

Mass of cooling water (m_w).

3. Corresponding to a condensation temperature of 35°C,

Ideal pressure of steam, = 0.05628 bar = = 42.32 mm of Hg

∴ Ideal vacuum = Barometer reading – Ideal pressure

= 750 – 42.32 = 707.68 mm of Hg

Vacuum efficiency, η_{vac} =

Example 8.23

The following observations were recorded during a test on a steam condenser:

Barometer reading = 765 mm of Hg

Condenser vacuum = 710 mm of Hg

Mean condenser temperature = 35°C

Condensate temperature = 28°C

Condensate collected per hour = 2 tonnes

Quantity of cooling water per hour = 60 tonnes

Temperature of cooling water at inlet = 10°C

Temperature of cooling water at outlet = 25°C

Find: (a) vacuum corrected to the standard barometer reading, (b) vacuum efficiency of the condenser, (c) undercooling of the condensate, (d) condenser efficiency, (e) quality of the steam entering of condenser,

(f) mass of air per m^3 of condenser volume and (g) mass of air per kg of uncondensed steam.

Solution

Given that Barometer reading = 765 mm of Hg; Condenser vacuum = 710 mm of Hg; T = 35°C = 35 + 273 = 308 K; t_c = 28°C; $\dot{m}_s = 2\ t/h$ = 2000 kg/h; $\dot{m}_w$ = 60 t/h = 60000 kg/h; t_i = 10°C; t_o = 25°C

1. Absolute pressure in the condenser = Barometer reading – Condenser vacuum

   = 765 – 710 = 55 mm of Hg

   Vacuum corrected to the standard barometer reading (assuming 760 mm of Hg)

   = 760 – 55 = 705 mm of Hg
2. From steam tables, corresponding to the mean condenser temperature of 35°C, we find that ideal pressure of steam,

   Ideal vacuum = Barometer pressure – Ideal pressure

$= 765 - 42.32 = 722.68$ mm of Hg

Vacuum efficiency, $\eta_{vac} = = 0.9824$ or 98.24%

3. Undercooling of the condensate = Mean condenser temperature − Condensate temperature

$$= 35 - 28 = 7°C$$

4. Pressure in the condenser, $p_c = 765 - 710 = 55$ mm of Hg

$$= 55 \times 0.00133 = 0.073 \text{ bar}$$

From steam tables, corresponding to a pressure of 0.073 bar, we find that vacuum temperature,

5. Let x = Quality of steam entering the condenser.

From steam tables, corresponding to a pressure of 0.073 bar, we find that,

$$h_f = 166.7 \text{ kJ/kg; and hfg} = 2407.4 \text{ kJ/kg}$$

Corresponding to a condensate temperature of 28°C, heat in condensate,

$$h_{f1} = 117.3 \text{ kJ/kg}$$

Total heat of entering steam,

$$h = h_f + xh_{fg} = 166.7 + x \times 2407.4$$

Mass of cooling water (m_w),

6. Absolute pressure of air (as per Dalton's law),

$p_a = p_c - p_s = 0.073 - 0.0562 = 0.0168$ bar

$= 0.0168 \times 10^5 = 1680$ N/m^2

∴ Mass of air per m^3 of condenser volume,

7. From steam tables, corresponding to 35°C (i.e., mean condenser temperature), specific volume of steam,

$$v_g = 25.245 \text{ m3/kg}$$

Air associated with 1 kg of steam at 35°C will occupy the same volume, i.e., 25.245 m^3

∴ Mass of air per kg of uncondensed steam,

Example 8.24

The following observations refer to a surface condenser:

Mass flow rate of condensate = 20 kg/min

Mass flow rate of cooling water = 800 kg/min

Mean temperature of condensation = 35°C

Condenser vacuum = 0.954 bar

Barometer reading = 1.03248 bar

Intel cooling water temperature = 20°C

Outlet cooling water temperature = 30°C

Temperature of the hot well = 29°C

Calculate the following:

1. Weight of air per unit volume of condenser
2. Entering condition of steam to the condenser
3. Vacuum efficiency of the condenser.

Properties of saturated steam:

Take *R* for air = 0.287 kJ/kg. K.

[IAS, 2009]

Solution

Given that $\dot{m}_s$ = 20 kg/min, $\dot{m}_w$ = 800 kg/min, T = 273 + 35 = 308 K, t_{wi} = 20°C, t_{wo} = 30°C, p_h = 1.03248 N/m^2, p_{cu} = 0.954 N/m^2, t_{hw} = 29°C

Absolute pressure in condenser, $p_t = (p_h - p_{cu})$ = 1.03248 – 0.954 = 0.07848 bar

At t = 35°C, p_s = 0.05628

Partial pressure of air, $p_a = p_t - p_s$ = 0.07848 – 0.05628 = 0.0222 bar

1. Mass of air, m_a =
2. At p_t = 0.07848 bar, h_f = 169.15 kJ/kg, h_{fg} = 2405.80 kJ.kg
3. Vacuum efficiency, η_{vac} =

Summary for Quick Revision

1. A condenser is a closed vessel in which steam is condensed by abstracting heat and the pressure is maintained below atmospheric pressure.
2. Steam condensers are basically of two types—jet condensers and surface condensers.
3. In jet condensers, the steam to be condensed mixes with the cooling water.
4. In surface condensers, steam does not mix with the cooling water.
5. Jet condensers can be classified as low-level counter flow or parallel flow type; high-level type, and ejector type.
6. Surface condensers can be classified as water tube type, central flow or regenerative type, and evaporative type.
7. Vacuum is sub-atmospheric pressure. It is measured by a vacuum gauge.
8. Absolute condenser pressure. $p = 0.133322\ (H_b - H_g)$ bar

   where H_b = actual barometric height, cm of Hg

   H_g = actual vacuum gauge reading, cm of Hg.

   Corrected or standard vacuum = $76 - (H_b - H_g)$ cm of Hg.

9. According to Delton's law of partial pressures,

   Total pressure in the condenser,

   p_t = absolute pressure of air, p_a + saturation pressure of steam, p_s

10. Mass of mixture,

    where m_a, m_g = mass of air and water vapour respectively.

v_a, v_g = specific volume of air and water vapour respectively.

11. Mass of cooling water required, for jet condensers
12. An Edward's air pump is used to maintain the desired vacuum in the condenser.
13. Vacuum efficiency is defined as the ratio of actual vacuum to ideal vacuum.

where p_b = barometric pressure

p_a = partial pressure of air

p_s = saturation pressure of steam

14. Condenser efficiency,

where t_{iw}, t_{ow} = inlet and outlet temperatures of cooling water

t_s = steam temperature in the condenser

15. A cooling tower is an artificial device used to cool hot cooling water coming out from the condenser.

Multiple-choice Questions

1. In a steam condenser, the partial pressure of steam and air are 0.060 bar and 0.007 bar, respectively. The condenser pressure is
    1. 0.067 bar
    2. 0.060 bar
    3. 0.053 bar
    4. 0.007 bar
2. Cooling tower in a steam power plant is a device for
    1. condensing steam into water
    2. cooling the exhaust gases coming out of the boiler

3. reducing the temperature of superheated steam
4. reducing the temperature of cooling water used in condenser

3. The function of the surface condenser is to
    1. lower the engine thermal efficiency
    2. lncrease the engine thermal efficiency
    3. lncrease the back pressure of the engine
    4. cool the exhaust gases
4. In a jet condenser,
    1. steam and cooling water mix together
    2. steam and cooling water do not mix together
    3. steam passes through tubes and cooling water surrounds them
    4. cooling water passes through tubes and steam surrounds them
5. A surface condenser is a
    1. water tube device
    2. steam tube device
    3. steam and cooling water mix to give the condensate
    4. All of the above
6. The air removal from the surface condenser leads to
    1. fall in condenser pressure
    2. rise in condenser pressure
    3. no change in condenser pressure
    4. rise in condenser temperature
7. Edward's air pump
    1. removes only air from the condenser
    2. removes air and vapour form the condenser
    3. removes only uncondensed vapour from condenser
    4. removes air along with vapour and condensed water from the condenser
8. In a surface condenser used in a steam power station, undercooling of condensate is undesirable as this would
    1. not absorb the gases in steam.
    2. reduce efficiency of the plant.
    3. increase the cooling water requirements.
    4. increase thermal stresses in the condenser
9. Consider the following statement:

    The effect of fouling in a water-cooled steam condenser is that it

1. reduces the heat transfer coefficient of water
2. reduces the overall heat transfer coefficient
3. reduces the area available for heat transfer
4. increases the pressure drop of water

Of these statements:

1. 1, 2, and 4 are correct
2. 2 and 4 are correct
3. 2 and 4 are correct
4. 1 and 3 are correct.

Review Questions

1. Define a condenser.
2. What are the functions of a condenser?
3. Name the various elements of a steam condensing plant.
4. What are the various types of steam condensers?
5. Differentiate between jet and surface condensers.
6. What is an evaporative condenser?
7. What are the requirements of a modern surface condenser?
8. List the advantages and disadvantages of a jet condenser.
9. What are the advantages and disadvantages of surface condensers?
10. State Dalton's law of partial pressures.
11. How the mass of cooling water can be estimated in a surface condenser?
12. What are the sources of air infiltration in a condenser?
13. What are the effects of air infiltration in condensers?
14. Define vacuum efficiency and condenser efficiency.
15. What is the role of cooling towers in surface condenser?

Exercises

8.1 The vacuum in a condenser handling 9000 kg/h of steam is found to be 72 cm

of Hg when the barometer reading is 76 cm of Hg and the temperature is 25°C. The air leakage amounts to 2 kg for every 5000 kg of steam. Determine the capacity of a suitable dry air pump in m^3/min required for the condenser. Take volumetric efficiency of the pump as 85%.

[**Ans.** 2.83m^3/min]

8.2 A steam condenser has separate air and condensate pumps. The entry to the air pump suction is screened. Steam enters the condenser at 38°C and the condensate is removed at 37°C. The air removed has a temperature of 36°C. If the quantity of air infiltration from various sources is 5 kg/h, determine the volume of air handled by the air pump

per hour. Compare this with the quantity that would have to be dealt with by using a combined air and condensate pump. Neglect the pressure due to air at entry of steam and assume uniform pressure in the condenser.

[**Ans.** 647.3m^3/h, 1267.38 m^3/h]

8.3 A surface condenser deals with 13625 kg of steam per hour at a pressure of 0.09 bar. The steam enters 0.85 dryness fraction and the temperature at the condensate and air extraction pipes is 36°C. The air leakage amounts to 7.26 kg/hour. Determine the following:

1. The surface area required if the average heat transmission rate is 4 kJ/cm per second.
2. The cylinder diameter for the dry air pump, if it is to be single acting and running at 60 rpm with a stroke-bore ratio of 1.25 and volumetric efficiency of 85%.

[**Ans.** (a) 1956.57 cm^2, (b) 74 cm]

8.4 A surface condenser, having an absolute pressure of 0.10 bar, is supplied with cooling water at the rate of 40kg of steam condensed. The rise in the temperature of cooling water is 14°C. Find the dryness fraction of steam entering the condenser. The condensate leaves at 44°C.

[**Ans.** 0.98]

8.5 A steam turbine discharges 5000 kg/h of steam at 40°C and 0.85 dry. The air leakage in the condenser is 15 kg/h. The temperature at the suction of air pump is 32°C and temperature of condensate is 35°C. Find (a) vacuum gauge reading, (b) capacity of air pump, (c) loss of condensate in kg/h, and (d) quantity of cooling water if its temperature rise is

limited to 10°C.

[**Ans.**(a) 70.45 cm, (b) 502.5 m^3/h, (c) 17 kg/h, (d) 254 × 103 kg/h]

8.6 A steam turbine uses 50,000 kg/h of steam. The exhaust steam with dryness fraction 0.9 enters the condenser fitted with water extraction and air pumps. When the barometer reads 76 cm of Hg, vacuum of air pump suction is 72 cm of Hg and temperature is 32°C. The air leakage is estimated at 500 kg/h. Calculate (a) net capacity of air pump and (b) quantity of water circulated per minute if temperature rise is limited to 15°C.

[**Ans.** (a) 1291.83 m^3.min, (b) 27160 kg/min]

8.7 The temperature in a surface condenser is 40°C and the vacuum is 69 cm of Hg while the barometer reads 75

cm of Hg. Determine the partial pressure of steam and air and the mass of air present per kg of steam.

[**Ans.** 0.07375 bar, 0.00625 bar, 0.136 kg/kg vapour]

8.8 In a condenser, to check the leakage of air, the following procedure is adopted:

After running the plant to reach the steady conditions, the steam supply to the condenser and also the air and condensate pumps are shut down, thus completely isolating the condenser. The temperature and vacuum readings are noted at shut down and also after a period of 5 minutes. They are 39°C and 68.5 cm Hg; and 28°C and 27 cm Hg respectively. The barometer reads 75 cm Hg. The effective volume of condenser

is 1 m^3.Calculate (a) the quantity of air leakage into the condenser during the period of observation, and (b) the quantity of water vapour condensed during the period.

[**Ans.** 0.3523 kg, 0.0214 kg]

ANSWERS TO MULTIPLE-CHOICE QUESTIONS

1. a
2. d
3. b
4. a
5. a
6. a
7. d
8. c
9. b

Chapter 9

Gas Power Cycles

9.1 ❒ Introduction

The gas as the working fluid does not undergo any phase change. The engines working on the gas cycles may be either cyclic or non-cyclic. The working fluid is assumed to obey all laws of a perfect gas.

In most of the gas power cycles, the working fluid consists mainly of air. Therefore, it is convenient to analyse the gas power cycles by devising an idealized cycle known as Air Standard Cycle, in which the working fluid is

pure air.

The following assumptions are made in the air-standard cycles:

1. The air as the working fluid follows the perfect gas law: pV = mRT.
2. The working fluid is homogeneous throughout and no chemical reaction takes place.
3. The specific heats of air do not vary with temperature.
4. The mass of air in the cycle remains fixed.
5. The combustion process is replaced by an equivalent heat addition process.
6. The exhaust process is replaced by an equivalent heat rejection process.
7. All processes are internally reversible.

The thermal efficiency, η_{th}, of air standard cycle is given by

The thermal efficiency is also called air-standard efficiency (η_a)

Relative efficiency, (η_r)= Actual thermal efficiency/Air standard efficiency.

9.2 ❒ Piston-cylinder Arrangement

The piston-cylinder arrangement is shown in Fig. 9.1. The various definitions are:

Bore (*d*): is the cylinder or piston diameter.

Stroke (*L*): is the distance moved by the piston in one direction. It is equal to twice the crank radius.

Figure 9.1 *Piston-cylinder arrangement*

Top Dead centre (T.D.C.): is the extreme position of the piston near to the head of cylinder. It is also called inner dead centre (I.D.C) position

Bottom Dead centre (B.D.C.): is the extreme position of the piston opposite

to the head of
cylinder. It is also called outer dead centre (O.D.C) position

$$L = T.D.C. - B.D.C.$$

Clearance Volume (V_c): is the minimum volume of clearance between the cylinder head and the piston at T.D.C. position.

Swept Volume (V_s): is the maximum volume swept by the piston in moving from T.D.C. to B.D.C. position or vice versa.

Cylinder Volume (V): is the sum of clearance volume and swept volume

$$V = V_c + V_s$$

Compression ratio (r): is the ratio of

cylinder volume to the clearance volume. Thus

Clearance ratio (c): is the ratio of clearance volume to stroke volume. Thus

Mean effective pressure (p_m): is the average pressure which, if acted on the piston during the entire power or outward stroke, would produce the same work output as the network output for the actual cyclic process. Thus

Work done per cycle,

Work done per minute = $p_m L A n$

Where

n = number of strokes per minute.

= N, for two stroke engines.

= $N/2$, for four stroke engines.

N = rpm

where p_m is in kPa.

9.3 ❒ Carnot Cycle

The Carnot cycle consists of two reversible isotherms and two reversible adiabatics, as shown in Fig. 9.2. For one kg of gas ($T_1 > T_2$), we have

Figure 9.2 *Carnot cycle: (a) p-v diagram, (b) T-s diagram*

Heat supplied,

Heat rejected,

Net work,

9.4 ❐ Stirling Cycle

This cycle works on the principle of regeneration by using a regenerator within the engine itself which would store the rejected heat energy during heat rejection process and supply the same during heat addition process. The Stirling cycle consists of two reversible isotherms and two reversible isochors. The $p - v$ and $T - s$ diagrams are shown in Fig. 9.3. We note that heat addition process 2–3 and heat rejection process 4–1 cancel each other, since the energy taken from the regenerator is returned in the latter. An alternative interpretation is

to consider the regenerator as a part of the system so that the heat exchange with the surroundings involves only the source at temperature T_1 and the sink at temperature T_2. The heat supplied per cycle is, therefore, during the process from 3–4 and heat rejection is during the process 1–2. For one kg of ideal gas ($T_1 > T_2$), we have

Heat supplied,

Heat rejected,

Thermal efficiency,

But $v_4 = v_3$, $v_1 = v_2$ and hence

Generally there is loss of heat due to radiation and poor conductivity of gas.

If η_r is the efficiency of the regenerator, then

Heat supplied = $RT_1 \ln r + (1 - \eta_r) c_v (T_1 - T_2)$

Heat rejected = $RT_2 \ln r + (1 - \eta_r) c_v (T_1 - T_2)$

Figure 9.3 *Stirling cycle: (a) p-v diagram, (b) T-s diagram*

9.5 ❐ Ericsson Cycle

This cycle also works on the principle of regeneration. It consists of two reversible isothermals and two reversible isobars. The *p-v* and *T-s* diagram are shown in Fig. 9.4. The heat rejected during the heat rejection process 4 – 1 at constant pressure is stored in the regenerator and the same is

supplied during heat addition process 2 – 3. Assuming 100% efficiency of the regenerator, we have

Heat supplied,

Heat rejected,

Where

Figure 9.4 *Ericsson cycle: (a) p-v diagram, (b) T-s diagram*

9.6 ❒ Atkinson Cycle

This cycle has two reversible adiabatics (isentropics), one isobar and one isochore. The *p-v* and *T-s* diagrams are shown in Fig. 9.5. For one kg of the working fluid, we have

Heat supplied, $q_s = c_v (T_3 - T_2)$

Heat rejected, $q_r = c_p (T_4 - T_1)$

Net work done, $w_{net} = q_s - q_r = c_v (T_3 - T_2) - c_p (T_4 - T_1)$

Thermal efficiency,

where

Let compression ratio,

Explosion ratio,

From compression process 1 – 2, we have

For constant volume heat addition process 2 – 3, we have

Figure 9.5 *Atkinson cycle: (a) p-v diagram, (b) T-s diagram*

From expansion process 3–4, we have

Substituting the values of T_2, T_3 and T_4, we get

9.7 ❐ Otto Cycle (constant volume cycle)

This cycle is composed of four internally reversible processes, two adiabatic and two constant volume processes. The *p-v* and *T-s* diagrams are shown in Fig. 9.6. The various processes are:

Process 1–2: Isentropic compression.

Process 2–3: Constant volume heat addition.

Process 3–4: Isentropic expansion.

Process 4–1: Constant volume heat rejection.

This cycle is used for spark ignition (petrol) engines.

Consider 1 kg of air flowing through the cycle. Since the air in the cylinder acts as a closed system, from first law of thermodynamics for isentropic compression and expansion, we have

$$q - w = \Delta u$$

For constant volume heat supplied and rejection processes, since $w = 0$,

$$q = \Delta u = c_v \, \Delta T$$

Figure 9.6 *Otto cycle: (a) p-v diagram, (b) T-s diagram*

Heat supplied $\quad q_s = q_{2-3} = c_v(T_3 - T_2)$

Heat rejected $q_r = q_{4-1} = c_v\,(T_4 - T_1)$

Work done per cycle, $w_{\text{net}} = q_s - q_r = c_v(T_3 - T_2) - c_v(T_4 - T_1)$

Air standard (or thermal) efficiency =

where

where pressure ratio

Figure 9.7 *Otto cycle thermal efficiency v's compression ratio*

The air standard efficiency of Otto cycle depends on compression ratio only and increases as compression ratio increases (Fig. 9.7). In actual engine working on Otto cycle, the compression ratio varies from 5 to 8. This engine is used for spark ignition engines working on

petrol.

Mean Effective Pressure (m.e.p.). It may be defined as the ratio of work done to the displacement volume of piston.

Work done, $w = c_v[(T_3 - T_2) - (T_4 - T_1)]$

Displacement volume, $v_s = v_1 - v_2$

M.E.P.,

9.8 ❐ Diesel Cycle

This cycle is used for compression ignition internal combustion engines working on diesel oil. The *p-v* and *T-s* diagrams are shown in Fig. 9.8. It consists of four internally reversible processes two adiabatic, one constant pressure and one constant volume. The

various processes are:

Process 1–2: Isentropic compression of air.

Process 2–3: Heat addition at constant pressure.

Process 3–4: Isentropic expansion of air.

Process 4–1: Heat rejection at constant volume.

Considering 1 kg of air.

Heat supplied, $q_s = q_{2-3} = c_p\,(T_3 - T_2)$

Heat rejected, $q_r = q_{4-1} = c_v\,(T_4 - T_1)$

Work done per cycle, $w = q_s - q_r = c_p$

$(T_3 - T_2) - c_v (T_4 - T_1)$

Thermal efficiency,

Let = compression ratio

Now

Let = cut off ratio

Figure 9.8 *Diesel cycle: (a) p-v diagram, (b) T-s diagram*

The thermal efficiency of diesel cycle increases as compression ratio increases but decreases as cut-off ratio increases (Fig. 9.9). The thermal efficiency of diesel cycle is less than that of Otto cycle. The compression ratio for diesel

cycle varies from 14 to 18.

Mean effective pressure

Now

Figure 9.9 *Diesel cycle thermal efficiency v's compression ratio*

9.9 ❐ Dual Cycle

In the dual cycle, part of the heat is supplied at constant volume and the rest at constant pressure. This is also called mixed or limited pressure cycle. The *p*-*v* and *T*-*s* diagrams are shown in Fig. 9.10. The various processes are:

Process 1–2: Isentropic compression of air

Process 2–3: Constant volume heat

addition

Process 3–4: Constant pressure heat addition

Process 4–5: Isentropic expansion of air

Process 5–1: Constant volume heat rejection

Considering 1 kg of working fluid,

Heat supplied, $q_s = c_v (T_3 - T_2) + c_p (T_4 - T_3)$

Heat rejected, $q_r = c_v (T_5 - T_1)$

Work done per cycle, $w = q_s - q_r = c_v (T_3 - T_2) + c_p (T_4 - T_3) - c_v (T_5 - T_1)$

Let, = compression ratio

Then from process 1–2, we have

$$T_2 = T_1 r^{\gamma-1}$$

From process 2 – 3,

Figure 9.10 *Dual cycle: (a) p-v diagram, (b) T-s diagram*

From process 3 – 4,

From process 4 – 5,

The variation of thermal efficiency of dual cycle is shown in Fig. 9.11.

Mean effective pressure,

Now

Figure 9.11 *Variation of thermal efficiency of dual cycle with compression ratio*

9.10 ❐ Brayton Cycle

The air standard Brayton or Joule cycle is a constant pressure cycle used in gas turbine power plants. The *p-v* and *T-S* diagrams are shown in Fig 9.12. It consists of the following processes:

1–2: Isentropic compression in the compressor

2–3: Constant pressure heat addition.

3–4: Isentropic expansion of air

4–4: Constant pressure heat rejection

Consider 1 kg of working fluid

From first law of thermodynamics for steady flow, rejecting ΔKE and ΔPE, we have

$$\delta_q - \delta_w = dh$$

Heat added, $q_s = h_3 - h_2 = c_p\,(T_3 - T_2)$

Heat rejected $q_r = h_4 - h_1 = c_p\,(T_4 - T_1)$

Net work done by turbine, $w_{\text{net}} = q_s - q_r = c_p\,[(T_3 - T_2) - (T_4 - T_1)]$

Also work done by turbine, $w_t = h_3 - h_4 = c_p\,(T_3 - T_4)$

Work consumed by compressor, $w_c = h_2 - h_1 = c_p\,(T_2 - T_1)$

$$w_{\text{net}} = c_p\,[(T_3 - T_4) - (T_2 - T_1)] = c_p\,[(T_3 - T_2) - (T_4 - T_1)]$$

Thermal efficiency,

Figure 9.12 *Brayton (or Joule) cycle: (a) p-v diagram, (b) T-s diagram*

From isentropic compression process 1 – 2, we have

and from isentropic expansion process 3 – 4,

The variation of thermal efficiency *v's* pressure ratio is shown in Fig. 9.13. The thermal efficiency increases with increasing values of pressure ratio. This cycle is used in gas turbines.

w_t = work done by turbine

w_c = work supplied to compressor

Pressure ratio for maximum work,

$$W = mc_p\,[(T_3 - T_2) - (T_4 - T_1)]$$

$$= mc_p\,[(T_3 - T_4) - (T_2 - T_1)]$$

Figure 9.13 *Efficiency v's pressure ratio in simple Brayton cycle*

9.11 ❒ Comparison between Otto, Diesel, and Dual Cycles

The comparison parameters selected are:

1. Equal compression ratio and heat input.
2. Constant maximum pressure and heat input.
3. Constant maximum pressure and work output.
4. Constant maximum pressure and temperature.
5. Equal compression ratio and heat rejection.

Figure 9.14 *Comparison for equal compression ratio and heat input:*

(a) p-v diagram, (b) T-s diagram

1. **Equal compression ratio and heat input.** The *p-v* and *T-s* diagrams for the three cycles are shown in Fig.9.14. The cycles have been represented as follows:

That cycle will be more efficient which rejects the least amount of heat after expansion, because

Thus the order of efficiencies is:

2. **Constant maximum pressure and heat input.** The *p-v* and *T-s* diagrams for Otto and Diesel cycles are shown in Fig. 9.15 for constant maximum pressure and heat input

Area $a - 2 - 3 - b$ = Area $a - 2' - 3' - b'$ for same heat input.

Thus cycle will be more efficient that rejects the least amount of heat.

(Area $a - 1 - 4 - b$) < (Area $a - 1 - 4' - b'$)

Thus the Diesel cycle is more efficient than the Otto cycle. A similar argument will show that the dual cycle performance falls between the other two. Thus

Figure 9.15 *Comparison for equal maximum pressure and heat input: (a) p-v diagram, (b) T-s diagram*

3. **Constant maximum pressure and work output.** The *p-v* and *T-s* diagrams are shown in Fig. 9.15 for maximum pressure and output.

Area 1 – 2 – 3 – 4 = Area 1 – 2′ – 3′ – 4′ for same work output

The cycle will be more efficient which rejects least amount of heat.

(Area a – 1 – 4 – b) < (Area a – 1′ – 4′ – b′)

Hence the Diesel cycle is more efficient than the Otto cycle. Thus

4. **Constant maximum pressure and temperature.** The *p-v* and *T-s* diagrams are shown in Fig. 9.16 for same maximum pressure and temporal are.

Now

Figure 9.16 *Comparison for equal maximum pressure and temperature: (a) p-v diagram, (b) T-s diagram*

All the cycles reject equal amount of heat.

Thus cycle will be more efficient which has higher heat addition.

Hence, the Diesel cycle is more efficient than the

Otto cycle. Thus

5. **Equal compression ratio and heat rejection**. The *p-v* and *T-s* diagrams are shown in Fig. 9.17, for equal compression ratio and heat rejection

For same q_r, if q_s is more, η is more.

Figure 9.17 *Comparison for equal compression ratio and heat rejection: (a) p-v diagram, (b) T-s diagram*

Example 9.1

A Carnot cycle has lowest temperature and pressure of 20°C and 1 bar. The pressures are: 4 bar after isothermal compression; 12 bar after isentropic compression and 6 bar after isothermal heat addition. Calculate (a) the highest temperature in the cycle, (b) thermal efficiency of the cycle, (c) mean effective pressure, and power

developed with 150 cycles per minute.

Solution

Given: $p_4 = 1$ bar, $T_2 = 293$ K, $p_2 = 12$ bar, $p_1 = 4$ bar, $p_3 = 6$ bar (Refer to Fig. 9.2)

Stroke volume, $v_s = v_4 - v_2 = 0.8409 - 0.0958 = 0.7451$ m^3/kg

1. Highest temperature in the cycle,
2. Heat supplied,

   = 79.67 kJ/kg

   Heat rejected,

   = – 58.07 kJ/kg

   Net work done, $w_{net} = q_s - q_r = 79.67 - 58.07 = 21.60$ kJ/kg

   Thermal efficiency,
3. Mean effective pressure =
4. Power developed = $w_{net} \times$ rpm/60 = 21.6 × 150/60 = 54 kW.

Example 9.2

The following particulars refer to an engine working on Stirling cycle.

Compression ratio = 3, lower temperature = 30°C, speed = 50 rpm, higher temperature = 550°C, regenerative efficiency =90%, initial pressure = 1 bar, heat added = 2300 kJ/min.

Calculate (a) work done per kg, (b) thermal efficiency, (c) mean effective pressure, and (d) indicated power.

Solution

Refer to Fig. 9.18.

1. Work done = $R(T_1 - T_2)$ ln r

= 0.287 (823 – 300) ln 3

= 163.956 kJ/kg

Heat added = $RT_1 \ln r + (1 - \eta_{reg})\, c_v(T_1 - T_2)$

= 0.287 × 823 × ln 3 + (1 – 0.9) × 0.718 × (823 – 303)

= 296.829 kJ/kg

2. Thermal efficiency
3. Air flow rate
4. Working cycles/min =

Mass of air per working cycle,

Figure 9.18

Stroke volume $V_s = V_1 - V_2$ = 0.2694 – 0.0898 = 0.1796m^3

Mean effective pressure,

5. Indicated power

Example 9.3

An engine working on Atkinson cycle takes in air at 1 bar and 25°C. The air is compressed isentropically by a compression ratio of 6:1. The heat is added at constant volume increasing the final pressure to 20 bar. Now, the air is expanded isentropically to 1 bar. The heat is rejected at constant pressure Calculate (a) pressure and temperature at various points, (b) work done per kg of air, and (c) cycle efficiency

Solution

Refer to Fig. 9.19.

1. $p_1 = 1$ bar, $T_1 = 25 + 273 = 298$ K

$$T_2 = T_1 r^{\gamma - 1} = 298 \times 6^{0.4} = 610.2 \text{ K}$$

$p_2 = p_1 r_\gamma = 1 \times 6^{1.4} = 12.286$ bar

$p_3 = 20$ bar

Heat added, $q_s = c_v(T_3 - T_2) = 0.718\ (993.3 - 610.2) = 279.4$ kJ/kg

Heat rejected, $q_r = c_p(T_4 - T_1) = 1.005\ (422.7 - 298) = 124.7$ kJ/kg

2. Work done, $w = q_s - q_r = 279.4 - 124.7 = 154.7$ kJ/kg
3. Thermal efficiency =

Figure 9.19

Example 9.4

In an engine working on ideal Otto cycle, the temperature at the beginning and end of compression are 45° C and 370°C. Find the compression ratio and air standard efficiency of the engine.

Assume $\gamma = 1.4$.

Solution

Refer to Fig. 9.6.

1. Compression, ratio,
2. Air standard efficiency,

Example 9.5

In an Otto cycle, air at 15°C and 1 bar is compressed adiabatically until the pressure is 15 bar. Heat is added at constant volume until the pressure rises to 40 bar. Calculate (a)the air-standard efficiency, (b) the compression ratio, and (c) the mean effective pressure for the cycle. Assume $c_v = 0.718$ kJ/kg. K, $\gamma = 1.4$ and $R = 8.314$ kJ/kmol.K.

Solution

The Otto cycle is shown in Fig. 9.20.

For isentropic process 1 – 2,

$$p_1 v_{1\gamma} = p v_{2\gamma}$$

Compression ratio,

Air standard efficiency,

Figure 9.20

For constant volume process 2 – 3:

Heat supplied $q_s = c_v(T_3 - T_2) = 0.718\ (1669.7 - 629.3) = 747.03$ kJ/kg

Work done, $w = \eta_a\, q_s = 0.539 \times 747.03 = 402.65$ kJ/kg

Mean effective pressure,

Specific volume, $v_1 - v_2 = 0.7064$ m^3/kg

Example 9.6

A gas engine working on Otto cycle has a cylinder of diameter 220 mm and stroke 300 mm. The clearance volume is 1600 cc. Find the air standard efficiency.

Assume $c_p = 1.004$ kJ/kg. K and $c_v = 0.718$ kJ/kg.K for air.

Solution

Swept volume,

Compression ratio,

Adiabatic index,

Air standard efficiency,

Example 9.7

The pressure limit in an Otto cycle are 1 bar and 20 bar. The compression ratio is 5. Calculate (a) thermal efficiency and (b) mean effective pressure. Assume $\gamma = 1.4$ for air.

Solution

1.

2. From Fig. 9.6, we have

$$p_2 = 1 \times 9.5182 = 9.5182 \text{ bar}$$

$$p_3 = 20 \text{ bar}$$

Pressure Ratio,

Example 9.8

A petrol engine with compression ratio of 5 develops 20kW indicated power and consumes 8 litres of fuel per hour. The specific gravity of fuel is 0.78 and its calorific value is 44 MJ/kg. Calculate the indicated thermal efficiency and relative efficiency. Take $= \gamma$ 1.4.

Solution

Air standard efficiency,

Fuel consumption = 8 × 0.78 × 1 = 6.24 kg/h

Indicated thermal efficiency,

Relative efficiency

Example 9.9

A diesel engine has a compression ratio of 20 and cut-off takes place at 5% of the stroke. Find the air-standard efficiency. Assume $\gamma = 1.4$.

Solution

Refer to Fig. 9.8.

Cut-off ratio,

Example 9.10

A diesel cycle operates at a pressure of 1 bar at the beginning of compression and the volume is compressed to th of the initial volume. Heat is supplied until the volume is twice that of the clearance volume. Calculate the mean effective pressure of the cycle. Assume $\gamma = 1.4$.

Solution

With reference to Fig. 9.8,

Swept volume, $v_s = v_1 - v_2 = (r - 1)$ $v_2 = (15 - 1)\ v_2 = 14\ v_2$

or

Process 1–2:

$$p_2 = p_3 = 49.31 \text{ bar}$$

$$= 0.071\ [49.31 + 122.55 - 79.27] = 6.685 \text{ bar}$$

Example 9.11

The mean effective pressure of an ideal diesel cycle is 10 bar. If the initial pressure is 1 bar and the compression ratio is 14, determine the cut off ratio and the air standard efficiency. Assume $\gamma = 1.4$.

Solution

See Fig. 9.8.

Work output

Work done per cycle

or $f(\rho) = \rho^{1.4} - 9.024\rho + 6.738 = 0$

We take $\rho = 2.64$

Example 9.12

In an engine working on the diesel cycle, the air-fuel ratio by weight is 50:1. The temperature of air at the

beginning of combustion is 40°C and the compression ratio is 19. What is the ideal efficiency of the engine. Calorific value of fuel is 42 MJ/kg. Assume $c_v = 0.717$ kJ/kgK, $c_p = 1.004$ kJ/kgK.

Solution

Refer to Fig. 9.8.

Process 1 – 2:

$$T_2 = 929.66 \text{ K}$$

Process 2 – 3:

Example 9.13

An air standard diesel cycle has a compression ratio of 16. The pressure at the beginning of compression stroke is 1 bar and the temperature is 20°C. The maximum temperature is 1430°C. Determine the thermal efficiency and the mean effective pressure for this cycle. Take $\gamma = 1.4$.

Solution

Refer to Fig. 9.8.

Process 2 – 3:

Example 9.14

An engine works on dual combustion cycle, the compression ratio being 11. The pressure at the commencement of combustion is 1 bar and the temperature is 90°C. The maximum pressure in the cycle is 50 bar and the constant pressure heat addition continues for of the stroke. Calculate the work done per kg of air and the ideal thermal efficiency. Assume, c_v = 0.718 kJ/kgK, c_p = 1.005 kJ/kgK.

Solution

$$T_1 = 90 + 273 = 363\text{K}$$

Refer to Fig. 9.21.

Figure 9.21

Process 1 – 2: $T_2 = T_1 r^{0.4} = 363 \times 11^{0.4} = 947.24$K

$$p_2 = p_1 r_\gamma = 1 \times 11 = 28.704 \text{ bar}$$

Process 2 – 3:

Process 3 – 4:

Process 4 – 5:

$$w = q_s - q_r = c_v (T_3 - T_2) + c_p (T_4 - T_3) - c_v (T_5 - T_1)$$
$$= [0.718(1650 - 947.24) + 1.005(2475 - 1650)] - 0.718 (1119.46 - 363)$$
$$= 1339.70 - 360.26 = 979.4 \text{ kJ/kg}$$

Example 9.21

A gas turbine operates on Brayton

cycle with in takes air at 1 bar and 15°C. The air is compressed to 15 bar and heated in a combustion chamber to 800°C. The hot air expands in the turbine to 1 bar. Find (a) the power developed, (b) thermal efficiency of the cycle. Take $\gamma = 1.4$ and $c_p = 1.005$ kJ/kgK.

Solution

Refer to Fig. 9.22

$$T_1 = 15 + 273 = 288\text{K}$$

Compressor work, $w_c = c_p (T_2 - T_1)$ = 1.005(456.14 - 288) = 168.98 kJ/kg

Turbine work, $w_t = h_3 - h_4 = c_p (T_3$

$- T_4)$

$$= 1.005(1073 - 677.47) = 397.5 \text{ kJ/kg}$$

Heat added, $q_s = q_{2\text{-}3} = h_3 - h_2 = c_p (T_3 - T_2)$

$$= 1.005\ (1073 - 456.14) = 619.94 \text{ kJ/kg}$$

Network, $w = w_t - w_c = 397.5 - 168.98 = 228.52$ kJ/kg

Power output = 228.52 kJ/kg

Example 9.15

Prove that the optimum pressure ratio r_p for maximum net-work done between the temperature T_1 and T_3 for a Brayton cycle is given by:

Solution

From Fig. 9.22, we have

Net-work done/kg of air,

$$w = (h_3 - h_2) - (h_4 - h_1) = c_p\,(T_3 - T_2) - c_p\,(T_4 - T_1)$$
$$= c_p\,(T_3 - T_4) - c_p\,(T_2 - T_1) = c_p\,T_3(1 - 1/x) - c_p\,T_1\,(x - 1)$$

or

Here y is constant and x is variable,

Example 9.16

An air-standard Brayton cycle has pressure ratio across the compression equal to 6. Air enters the compressor at 1 bar and 27°C.

The maximum temperature of the cycle is 850°C. Calculate the specific output of the cycle. What will be the power/developed by the unit for a mass flow rate of 10 kg/s. For air, $\gamma = 1.4$ and $c_p = 1.005$ kJ/kg.K.

Solution

From Fig. 9.12, we have

Specific output, $w = c_p\,(T_3 - T_2) - c_p(T_4 - T_1)$

$= 1.005\,[(1123 - 500.8) - (672.86 - 300)] = 250.58$ kJ/kg

Power developed, $P = mw = 10 \times 250.58 = 2505.8$ kW

Example 9.17

In an I.C. engine operating on the dual cycle (limited pressure cycle), the temperature of the working fluid (air) at the beginning of compression is 27°C. The ratio of the maximum and minimum pressures of the cycle is 70 and the compression ratio is 15. The amounts of heat added at constant volume and at constant pressure are equal. Compute the air standard thermal efficiency of the cycle.

[IES, 1993]

Solution

The *p*-*v* diagram for the dual cycle is shown in Fig. 9.23.

$$T_1 = 273 + 27 = 300 \text{ K}$$

Process 2 – 3:

Now $c_v (T_3 - T_2) = c_p (T_4 - T_3)$

Process 3 – 4:

Also $v_5 = v_1$

Process 4 – 5:

Figure 9.23 *Dual cycle on p-v diagram*

Thermal efficiency,

Example 9.18

In an air-standard Brayton cycle the minimum and maximum temperatures are 300 K and 1200 K, respectively. The pressure ratio is that which maximizes the network developed by the cycle per unit mass of air flow. Calculate the compressor and turbine work, each in kJ/kg air, and the thermal efficiency of the cycle.

[IES, 1994]

Solution

With reference to Fig. 9.25, T_{max} = 1200 K and T_{min} = 300 K

For maximum work, optimum pressure ratio,

Compressor work,

$$w_c = c_p\,(T_2 - T_1)$$
$$= 1.005(600 - 300) = 301.5 \text{ kJ/kg}$$

Turbine work, $w_t = c_p\,(T_3 - T_4) =$ 1.005(1200 – 600)= 603kJ/kg

Heat supplied, $Q_s = c_p\,(T_3 - T_2) =$ 1.005(1200 – 600) = 603kJ/kg

Thermal efficiency,

Figure 9.25 *Brayton cycle on T-s diagram*

Example 9.19

Derive an expression for air standard efficiency of the following cycle in terms of compression ratio R, c_v and γ.

1. an isothermal compression, compression ratio, *r*
2. an increase of pressure at constant volume.
3. an adiabatic expansion.

[IES, 1997]

Solution

The *p-v* diagram is shown in Fig. 9.28

Process 1 – 2: Isothermal compression

Process 2 – 3: Constant volume process

Process 3 – 1: Adiabatic expansion

$$W_{3-1} = mc_v\,(T_3 - T_1)$$
$$Q_{3-1} = 0$$

Air standard efficiency,

Figure 9.28 *p-v diagram*

Example 9.20

An air standard Otto cycle has a volumetric compression ratio of 6, the lowest cycle pressure of 0.1 MPa and operates between temperature limits of 27°C and 1569°C.

1. Calculate the temperature and pressure after the isentropic expansion (Ratio of specific heats = 1.4).
2. Since it is observed that values in (a) are well above the lowest cycle operating conditions, the expansion process was allowed to continue down to a pressure of 0.1 MPa. Which process is required to complete the cycle? Name, the cycle so obtained.
3. Determine by what percentage the cycle efficiency has been improved.

[GATE 1994]

Solution

The *p-v* diagram is shown in Fig. 9.29.

$$p_1 = 0.1\text{MPa}$$
$$T_1 = 27 + 273 = 300\text{K}$$
$$T_3 = 1569 + 273 = 1842\text{K}$$

1.

Figure 9.29 *Otto cycle*

2. Constant pressure scavenging is required to complete the cycle with expansion process to continue down to 0.1 MPa as shown in Fig. 9.30. The cycle is called Atkinson cycle.
3. Improvement in cycle efficiency = 59.25 – 51.16 = 8.09%

Figure 9.30 *Atkinson cycle*

Example 9.21

Air enters the compressor of a gas turbine plant operating on Brayton cycle at 1 bar, 27°C. The pressure ratio in the cycle is 6. If $W_t = 2.5\ W_c$ where W_t and W_c are the turbine and

compressor work respectively, calculate the maximum temperature and tin cycle efficiency.

[GATE, 1996]

Solution

The Brayton cycle is shown in Fig. 9.31

$$p_1 = 1\text{bar} = 1 \times 10^5 \text{ N/m}^2$$
$$T_1 = 27 + 273 = 300\text{K}$$

Figure 9.31 *Brayton cycle*

Cycle efficiency

Example 9.22

The minimum pressure and temperature in an Otto cycle are 100

kPa and 27°C. The amount of heat added to the air cycle is 1500 kJ/kg. Determine the pressure and temperatures at all points of the air standard Otto cycle. Also calculate the specific work and the thermal efficiency of the cycle for a compression ratio 8:1. (Take c_v (air) = 0.72 kJ/kg K and $(c_p/c_v)_{air}$ = 1.4).

[GATE, 1998]

Solution

The Otto cycle is shown in Fig. 9.32

Figure 9.32 *Otto cycle*

Specific work, $w_{net} = q_{2-3} - q_{4-1} =$ 1500 −652.91=847.09 kJ/kg

Example 9.23

A Brayton cycle (air standard) has a pressure ratio of 4 and inlet conditions of one standard atmospheric pressure and 27°C. Find the air flow rate required for 100 kW power output if the maximum temperature in the cycle is 1000°C. Assume $\gamma = 1.4$ and c_p = 1.0 kJ/kg K.

[**GATE, 2001**]

Figure 9.33 *Brayton cycle*

Solution

Refer to Fig. 9.33.

Example 9.24

Calculate the percentage loss in the ideal efficiency of a diesel engine with compression ratio 14 if the fuel cut-off is delayed from 5% to 8%.

[GATE, 2000]

Solution

The diesel cycle is shown in Fig. 9.34

Compression ratio, $r = 14$, Cut-off ratio, $\rho = 0.05$

Change in efficiency

Figure 9.34 *Diesel cycle*

Example 9.25

A reciprocating engine works on Otto cycle where the temperature at the beginning and end of the compression stroke are found to be 40°C and 360°C respectively. Assuming the ratio of specific heats at constant pressure and constant volume to be 1.4, determine the air standard efficiency and the compression ratio.

[IES 1984]

Solution

The *p-v* diagram is shown in Fig. 9.35

$$T_1 = 273 + 40 = 313\text{K}$$
$$T_2 = 273 + 360 = 633\text{K}$$

Process 1 – 2:

Compression ratio,

Air-standard efficiency

Figure 9.35 *p-v diagram for Otto cycle*

Example 9.26

An engine working on the ideal Otto cycle takes in air at 1 bar and 30°C which is compressed to 15 bar at the end of the compression stroke. The temperature attained at the end of constant volume heat addition is 900°C. Assuming adiabatic index to be 1.4, determine (a) the compression ratio, (b) the thermal

efficiency, (c) the temperature at the end of compression, (d) the pressure at the end of constant volume heating, and (e) the mean effective pressure.

[IES 1986]

Solution

The *p-v* diagrams are shown in Fig. 9.37.

$$p_1 = 1\text{bar}, T_1 = 273 + 30 = 303\text{K}$$
$$p_2 = 15\text{ bar}, T_3 = 273 + 900 = 1173\text{K}, \gamma = 1.4$$

1. Compression ratio,
2. Thermal efficiency,
3.
4.
5. Mean effective pressure,

Figure 9.37 *p – v diagram for Otto cycle*

Example 9.27

The diameter and stroke of a gas engine cylinder are 18 cm and 30 cm respectively. The ratio of expansion is 5. The pressure and temperature of the mixture at the beginning of compression are 1 bar and 100°C respectively. Find the index of the compression process and the weight of the mixture in the cylinder, if the pressure at the end of compression is 7 bar. Also calculate the work done and heat transferred during the process, indicating the direction of flow. Assume $R = 0.287$ kJ/kg K and ratio of specific heats equal to 1.4 for the mixture. Take 1 bar = 100 kPa.

[IES, 1987]

Solution

The p-V diagram is shown in Fig. 9.38

Figure 9.38

Work done

Heat added,

Now

– ve sign indicates that heat is rejected by the system.

Example 9.28

A four-stroke limited pressure cycle (diesel) engine draws 1.2 kg/s of air at 1 atm and 27°C. Compression ratio of the cycle is 16. Pressure ratio during constant volume heat addition is 2.0. Total heat added is equal to 2310 kJ/kg of air in the cylinder. Determine

1. pressure, volume and temperature at all salient points,
2. % of heat added during constant pressure process,
3. cut-off ratio,
4. thermal efficiency, and
5. mean effective pressure.

Represent the cycle on p-v and T-s planes.

Assume $c_p = 1.005$ kJ/kg.K and $c_v = 0.718$ kJ/kg.K

[IES. 1988]

Solution

The *p-v* and *T-s* diagrams are shown in Fig. 9.39.

1.

Figure 9.39 *Limited pressure cycle (diesel) 4-stroke engine: (a) p-v diagram, (b) T-s diagram*

Heat added,

2. Heat added during constant pressure process,

$$q_p = c_p (T_4 - T_3) = 1.005 (3467.64 - 1818.86) = 1657.024 \text{ kJ/kg}$$

Percentage of heat added =

3. Cut-off ratio,
4. Thermal efficiency,

Heat rejected during process 5 – 1

5. Mean effective pressure,

Example 9.29

A reversible cycle using an ideal gas as the working substance consists of an isentropic compression from an initial temperature to 555 K, a constant volume process from 555 K to 835 K, a reversible adiabatic expansion to 555 K, a constant pressure expansion from 555 K to 835 K, followed by constant volume process to the initial temperature. Draw the cycle on *p*-*v* amd *T*-*s* diagrams and determine the initial temperature. Take $\gamma = 1.40$. Also compute the work done.

[IES, 1989]

Solution

The p-v and T-s diagrams are shown in Fig 9.40 (a) and (b) respectively.

Process 1 – 2:

Process 3 – 4:

Process 4 – 5:

Now

Work done

Figure 9.40 *(a) p-v diagram, (b) T-s diagram*

Fill in the Blanks

1. The thermal efficiency of the Otto cycle_______as compression ratio increases.
2. The compression ratio of Otto cycle lies between_______.
3. The compression ratio is defined as the ratio of_______volume to_______volume.
4. The cylinder volume is the sum of_______volume and_______volume.
5. The Otto cycle is a_______cycle.

6. The heat addition and rejection in the Otto cycle take place at_______.
7. The heat addition in diesel cycle take place at constant_______.
8. The thermal efficiency of diesel cycle_______as cut off ratio increases.
9. The thermal efficiency of dual cycle equals that of Otto cycle at_______ equal to unity.
10. The thermal efficiency of dual cycle equals that of diesel cycle at_______equal to unity.
11. The heat addition and rejection in Brayton cycle take place at constant_______.
12. For equal compression ratio and heat input, thermal efficiency of Otto cycle is_______ than diesel and dual cycles.
13. For constant maximum pressure and heat input, thermal efficiency of diesel cycle is _______than dual and Otto cycles.
14. A cycle is said to be more efficient which rejects_______amount of heat.

Answers

1. increases
2. 5 to 8
3. cylinder, clearance
4. clearance, swept
5. constant volume
6. constant volume
7. pressure
8. decreases
9. cut-off ratio
10. pressure ratio
11. pressure
12. more
13. more
14. least.

True or False

State true (T) or false (F)

1. Displacement volume is the volume swept by the piston.
2. The Otto cycle is a constant volume cycle.
3. The compression ratio of Otto cycle varies from 5 to 8.
4. The Otto cycle is used in diesel engine.
5. The thermal efficiency of Otto cycle depends on compression ratio only.
6. The thermal efficiency of Otto cycle decreases as compression ratio increases.
7. The heat addition in diesel cycle is at constant pressure.
8. The cut-off ratio of a diesel engine is of the order of 2.
9. The thermal efficiency of diesel engine increases as cut-off ratio increases.
10. The thermal efficiency of a dual cycle lies between Otto and diesel cycle.
11. Joule cycle is used in a gas turbine.
12. Brayton cycle is a constant pressure cycle.

Answers

1. T
2. T
3. T
4. F
5. T
6. F
7. T
8. T
9. F
10. T
11. T
12. T

Multiple-choice Questions

1. The Otto cycle is also known as:
    1. constant pressure cycle
    2. constant temperature cycle
    3. constant volume cycle
    4. constant entropy cycle
2. The efficiency of Diesel cycle approaches that of Otto cycle when cut-off is:

    1. increased
    2. decreased
    3. constant
    4. zero
3. For the same maximum pressure and temperature of the Otto, Diesel and Dual air standard cycles, what will be same for the engine?
    1. compression ratio
    2. heal supplied
    3. thermal efficiency
    4. heat rejected
4. In a four stroke engine, a working cycle is completed in following revolutions of crankshaft:
    1. one
    2. two
    3. three
    4. four.
5. The air standard cycle on which the petrol engine works, is:
    1. Otto cycle
    2. Carnot cycle
    3. Joule cycle
    4. Dual cycle
6. The compression ratio in a petrol engine is of the order of :
    1. 5 to 8
    2. 10 to 15
    3. 15 to 25
    4. 25 to 30
7. A carburetor mixes in a petrol engine
    1. petrol and air
    2. diesel and air
    3. petrol and lubricating oil
    4. petrol, lubricating oil and air
8. The compression ratio in a diesel engine is of the order of
    1. 5 to 8
    2. 10 to 15
    3. 15 to 25
    4. 25 to 30
9. Which of the following does not relate to spark ignition engine?
    1. ignition coil
    2. spark plug
    3. distributor
    4. fuel injector

10. In a four stroke cycle diesel engine, during suction stroke
    1. only air is sucked
    2. only fuel is sucked
    3. mixture of fuel and air is sucked
    4. mixture of air and lubricating oil is sucked
11. The thermal efficiency of petrol engine as compared to diesel engine is:
    1. higher
    2. lower
    3. same for same power output
    4. same for same speed
12. The air standard Otto cycle consists of
    1. two constant pressure and two constant entropy processes
    2. two constant volume and two constant entropy processes
    3. two constant pressure and two constant volume processes
    4. two constant pressure and two constant temperature processes
13. The variation of work output *v's* pressure ratio for Otto cycle with fixed compression ratio and adiabatic index is shown below. The correct choice is:
    1. A
    2. B
    3. C
    4. D
14. The variation of thermal efficiency *v's* compression ratio for Otto cycle with fixed value of pressure ratio and adiabatic index is shown below. The correct choice is:
    1. A
    2. B
    3. C
    4. D
15. In a diesel cycle, the thermal efficiency for a fixed compression ratio and adiabatic index
    1. increases as cut-off ratio increases
    2. decreases as cut-off ratio increases
    3. remains same with increase in cut-off ratio
    4. none of the above.
16. Match the following

Air-standard cycle

1. Otto cycle
2. Diesel cycle
3. Carnot cycle
4. Joule cycle.

p-v diagram

1.
2.
3.
4.

Codes:

A B C D

1. 2 3 1 4
2. 2 3 4 1
3. 3 2 4 1
4. 3 2 1 4

17. Match the following:

Air standard cycle

1. Otto cycle
2. Diesel cycle
3. Carnot cycle
4. Brayton cycle

T-s diagram

1.
2.
3.
4.

Codes:

A B C D

1. 3 1 2 4
2. 3 1 4 2
3. 1 3 4 2
4. 1 3 2 4

18. Match the following:

p-v diagrams

1.
2.
3.
4.

T-s diagrams

1.
2.
3.
4.

Codes:

A B C D

1. 4 3 2 1
2. 4 3 1 2
3. 3 4 1 2
4. 3 4 2 1

19. A system comprising of a pure substance executes reversibly a cycle 1 – 2 – 3 – 4 –1 consisting of two isentropic and two isochoric process as shown in figure below:

Which one of the following is the correct representation of this cycle on the temperature-entropy coordinates?

1.

2.
3.
4.

20. Consider the following four cycles on *T-s* plane:
    1.
    2.
    3.
    4.

The correct sequence is:

1. Otto, Diesel, Carnot, Joule
2. Otto, Diesel, Joule, Carnot
3. Diesel, Otto, Carnot, Joule
4. Diesel, Otto, Joule, Carnot

21. In an air-standard Otto cycle, r is the compression ratio and γ is an adiabatic index, the air standard efficiency is given by
    1. $\eta = 1 - 1/r^{(\gamma-1)}$
    2. $\eta = 1 - 1/r^{\gamma}$
    3. $\eta = 1 - 1/r^{(\gamma-1)/\gamma}$
    4. $\eta = 1 - 1/r^{(\gamma-1)/2\gamma}$
22. The order of values of thermal efficiency of Otto, Diesel, and Dual cycles, when they have equal compression ratio and heat rejection, ratio is given by
    1. $\eta_{\text{Otto}} > \eta_{\text{diesel}} > \eta_{\text{dual}}$
    2. $\eta_{\text{diesel}} > \eta_{\text{dual}} > \eta_{\text{Otto}}$
    3. $\eta_{\text{dual}} > \eta_{\text{diesel}} > \eta_{\text{Otto}}$
    4. $\eta_{\text{Otto}} > \eta_{\text{dual}} > \eta_{\text{diesel}}$
23. Match List I with List II and select the correct answer using the codes given below the lists:

Codes:

	A	B	C	D
1.	3	4	1	2
2.	1	4	3	2
3.	3	2	1	4
4.	1	2	3	4

24. Match List I with List II and select the correct answer using the codes given below the lists:

Codes:

A B C D

1. 3 4 1 2
2. 1 4 3 2
3. 3 2 1 4
4. 1 2 3 4

25. Otto cycle efficiency is higher than Diesel cycle efficiency for the same compression and heat input because, in Otto cycle
    1. combustion is at constant volume
    2. expansion and compression air isentropic
    3. maximum temperature is higher
    4. Heat rejection is lower
26. Which one of the following cycles represent Brayton cycle?
    1.
    2.
    3.
    4.
27. *T*-*s* diagram for an air-standard cycle is shown below

    The same cycle on *p*-*V* diagram will be represented as

    1.
    2.
    3.
    4.
28. Consider the following statements regarding Otto cycle
    1. It is not a reversible process.
    2. The efficiency can be improved by using a working fluid of higher value of ratio of specific heats.
    3. The practical way of increasing its efficiency is to increase the compression ratio.
    4. Carbureted gasoline engines working on Otto cycle

can work with compression ratios more than 12.

Of these statements:

1. 1, 3, and 4 are correct
2. 1, 2 and 3 are correct
3. 1, 2 and 4 are correct
4. 2, 3 and 4 are correct

29. For the same maximum pressure and heat input, the most efficient cycle is
    1. Otto cycle
    2. Diesel cycle
    3. Brayton cycle
    4. Dual combustion cycle
30. Atkinson gas cycle has a special feature that
    1. isentropic compression and expansion are of equal volume
    2. isentropic compression is smaller than expension
    3. isentropic compression is longer than expansion
    4. it employs shortest compression and expansion
31. Ericsson cycle consists of the following four processes
    1. two isothermals and two isentropics
    2. two isothermals and two constant volumes
    3. two isothermals and two constant pressures
    4. two adiabatics and two constant pressures
32. Match List I with List II anil select the correct answers using the codes given below the lists

Codes:

	A	B	C	D
1.	3	4	2	1
2.	3	4	1	2
3.	4	3	1	2
4.	4	3	2	1

33. Match List I (details of process of the cycle) with List II (name of the cycle) and select the correct answer using the codes given below the lists

Codes:

A B C D

1. 4 3 1 2
2. 4 3 2 1
3. 3 4 1 2
4. 3 4 2 1

34. For maximum specific output of a constant volume cycle (Otto cycle)
    1. the working fluid should be air
    2. the speed should be high
    3. suction temperature should be high
    4. temperature of the working fluid at the end of compression and expansion should be equal
35. For a Brayton cycle, match the following using codes given below:

Codes:

A B C D

1. 1 2 3 4
2. 4 3 2 1
3. 2 3 4 1
4. 3 4 1 2

36. For the same maximum pressure and temperature
    1. Otto cycle is more efficient than diesel cycle
    2. Diesel cycle is more efficient than Otto cycle
    3. Dual cycle is more efficient than Otto and diesel cycle
    4. Dual cycle is less efficient than Otto diesel cycles.
37. A Otto cycle on internal energy (U) and entropy (*s*) diagram is shown in
    1. Fig. (a)
    2. Fig. (c)
    3. Fig. (b)

4. Fig. (d)

38. For constant maximum pressure and heat input, the air standard efficiency of gas power cycle is in the order
    1. Diesel cycle, dual cycle, Otto cycle
    2. Otto cycle, Diesel, cycle, dual cycle
    3. Dual cycle, Otto cycle, Diesel cycle
    4. Diesel cycle. Otto cycle, dual cycle
39. The use of regenerator in a gas turbine cycle
    1. increases efficiency but has no effect on output
    2. increase output but has no effect on efficiency
    3. increase both efficiency and output
    4. increases efficiency but decreases output
40. Consider the following statements with reference to gas turbine cycle

    1. Regeneration increases thermal efficiency
    2. Reheating decreases thermal efficiency
    3. Cycle efficiency increases when maximum temperature of the cycle is increased

Of these statements

1. 1, 2 and 3 are correct
2. 2 and 3 are correct
3. 1 and 3 are correct
4. 1 and 3 are correct

41. Match figures of Column 1 with those given in Column II and select the correct answer using the codes given below the columns:

Column I

p-v diagrams

1.
2.
3.

Column II

T-s diagrams

1.
2.
3.

Codes:

	A	B	C
1.	1	2	3
2.	2	3	1
3.	3	1	2
4.	3	2	1

42. Which one of the following diagrams represents Otto cycle on temperature (T)-entropy (s) plane?
    1.
    2.
    3.
    4.
43. Which one of the following p-T diagrams illustrates the Otto cycle of an ideal gas?
    1.
    2.
    3.
    4.
44. Match List I with List II and select the correct answer using the codes given below the lists

Codes:

	A	B	C	D
1.	3	1	4	2
2.	5	4	3	2
3.	2	3	4	5
4.	1	5	2	3

45. For a heat engine operating on Carnot cycle the work output is th of the heat transferred to the cold system. The efficiency of the engine is
    1. 20%
    2. 25%
    3. 25%
    4. 20%
46. Match List I with List II

Codes:

	A	B	C	D
1.	3	1	4	2
2.	5	4	3	2
3.	2	3	4	5
4.	1	5	2	3

47. With reference to air standard Otto and Diesel cycles, which of the following statements are true?
    1. For a given compression ratio and the same state of air before compression. Diesel cycle is less efficient than an Otto cycle
    2. For a given compression ratio and the same state of air before compression, Diesel cycle is more efficient than an Otto cycle
    3. The efficiency of a Diesel cycle decreases with an increase in the cut-off ratio
    4. The efficiency of a diesel cycle increases with an increase in the cut-off ratio
48. For the same maximum pressure and heat input, the most efficient cycle is
    1. Diesel cycle
    2. Dual cycle
    3. Otto cycle
    4. Stirling cycle
49. An Otto cycle operates with volumes of 40 cm^3 and 400 cm^3 at top dead centre (TDC) and bottom dead centre (BDC) respectively. If the power output is 100 kW, what is heat input, in kJ/s? Assume $\gamma = 1.4$

1. 166
2. 145
3. 110
4. 93

50. A cycle consisting of two reversible isothermal process and two reversible isobaric processes is known as
    1. Atkinson cycle
    2. Stirling cycle
    3. Brayton cycle
    4. Ericsson cycle
51. The air standard diesel cycle is less efficient than the Otto cycle for the
    1. same compression ratio and heat addition
    2. same pressure and heat addition
    3. same rpm and cylinder dimensions
    4. same pressure and compression ratio.
52. A pressure-volume diagram for some engine is shown in figure *A*. This diagram has been, closely translated into temperature-entropy diagram in figure *B*. The point *X* shown on *p-v* diagram is represented on *T-s* diagram by
    1. points *S*
    2. point *P*
    3. point *Q*
    4. point *R*

Review Questions

1. What do you mean by air-standard cycle? What are the assumptions for air-standard cycle?
2. Define thermal efficiency and relative efficiency of an air cycle.
3. Differentiate between cylinder volume and swept volume.
4. Define compression ratio and clearance ratio.
5. What is mean effective pressure?
6. Write the formula for the thermal efficiency of the Otto cycle.
7. What are the processes involved in the Otto cycle?
8. Draw the *p-v* and *T-S* diagrams for the Otto cycle.
9. Write the formula for the thermal efficiency of Diesel cycle.
10. Draw the variation of thermal efficiency against compression ratio of an Otto cycle.
11. How does the efficiency of Diesel cycle vary with cut-off ratio?
12. For what type of engines the diesel cycle is used?
13. What are the processes involved in a diesel cycle?

14. What is a dual cycle?
15. What are the processes involved in a Joule cycle?
16. What are the applications of Brayton cycle?
17. Compare the Otto, Diesel and Brayton cycles with respect to heat addition and heat rejection.

Exercises

9.1 In an air-standard Otto cycle the compression ratio is 10. The inlet temperature and pressure is 37.8°C and 1 bar. The maximum temperature of the cycle is 1060°C. Calculate heat supplied per kg of air, work done per kg of air, maximum pressure of cycle, and thermal efficiency.

[**Ans.** 396.55 kJ/kg, 238.68 kJ/kg, 42.89 kJ/kg, 60.18%]

9.2 The compression ratio of an Otto cycle is 8. The pressure and temperature at the beginning of compression stroke are 1 bar and 26.7°C. The heat added to the air per cycle is 2079 kJ/kg of air.

Calculate the pressure and temperature at the end of each process of the cycle. Thermal efficiency, and mean effective pressure.

[**Ans.** 688.5 K; 18.38 bar; 3584 K; 99.678 bar; 1560 K; 9.20 bar; 56.47%; 19.568 bar]

9.3 The compression ratio of an Otto cycle is 10 and the suction conditions are 1 bar and 50°C. If the heat rejection equals 3517 kJ/kg, calculate the air-standard efficiency and work ratio.

[**Ans.** 60.18%, 0.938]

9.4 An air-standard diesel cycle has a compression ratio of 16 and a cut-off ratio of 2. At begining of compression the cylinder volume is 0.001415 m^3, the air pressure is 1 bar and the temperature is 49.5°C. Calculate the thermal

efficiency.

[**Ans.** 61.38%]

9.5 A diesel engine with compression ratio of 15 is operating with a fuel having a heating value of 44505 kJ/kg. At half load this engine requires a 70 to 1 ratio of air to fuel.

1. What value of cut-off ratio should be used if $\gamma = 1.4$ and temperature at the beginning of compression is 40°C?
2. What is the ideal cycle efficiency?

[**Ans.** 1.68, 62%]

9.6 Calculate the thermal efficiency and mean effective pressure of a dual cycle having compression ratio equal to 10 when the minimum temperature is 2000°C and the maximum pressure is 70 bar. The pressure and temperature at the start of compression are 1 bar and 17°C respectively.

[**Ans.** 60%, 9.439 bar]

9.7 An engine operates on diesel cycle with a compression ratio of 12. The fuel is injected for 10% of stroke. The pressure of air entering the cylinder is 0.98 bar and its temperature is l5°C. Calculate cut-off ratio temperature at the end of compression process, and heat input.

[**Ans.** 2.1,778.1 K. 860.3 kJ/kg]

9.8 The compression ratio of an engine operating on dual cycle is 8. The diameter of cylinder is 25 cm and stroke is 30 cm. At the start of compression the air temperature is 21.6°C and the pressure is 56 bar. If heat is added at constant pressure during 3% of stroke, calculate a net work of the cycle, and

thermal efficiency.

[**Ans.** 768.6 kJ/kg, 59.93%]

9.9 In an air-standard Brayton cycle, the air enters the compressor at 1 bar and 25°C. The pressure after compression is 3 bar. The temperature at turbine inlet is 650°C. Calculate per kg of air heat supplied, heat rejected, work available at the shaft, temperature of air leaving the turbine, and cycle efficiency.

[**Ans.** 519.6 kJ/kg, 378 kJ/kg, 139.5 kJ/kg, 674 K, 26.97%]

9.10 In a Brayton cycle, air enters the compressor at 1 bar and 15°C and leaves the combustion chamber at 1400 K. Calculate the thermal efficiency.

[**Ans.** 48.20%]

9.11 An engine working on the Otto

cycle is supplied with air at 0.1 MPa and 35°C. The compression ratio is 8. Heat supplied is 2100 kJ/kg. Calculate the maximum pressure and temperature of the cycle, cycle efficiency, and the mean effective pressure. For air c_p = 1.005 kJ/kg.K, c_v = 0.718 kJ/kg.K and R = 0.287 kJ/kg.K.

[**Ans.** 9.435 MPa, 3639.4 K, 56 5%, 1.533MPa]

9.12 In an air-standard diesel cycle, the compression ratio is 16. At the beginning of isontropic compression, the temperature is 15°C and the pressure is 0.1 MPa. The temperature at the end of the constant pressure process is 1480°C. Calculate the cut-off ratio, heat supplied per kg of air, the cycle efficiency, and the m.e.p.

[**Ans.** 2.01, 889.4 k.J/kg. 76 43%, 871.766 kPa]

9.13 An air standard dual cycle has a compression ratio of 16. The compression begins at 1 bar and 50°C. The maximum pressure is 70 bar. The heat transferred to air at constant pressure is equal to that at constant volume. Calculate cycle efficiency, and m.e.p of cycle c_p= 1.005 kJ/kg.K, c_v = 0.718 kJ/kg/K.

[**Ans.** 66.31%, 9.752 bar]

9.14 An internal combustion engine works on diesel cycle with a compression ratio of 14 and cut-off takes place at 10% of the stroke. Find the ratio of cut-off and the air-standard efficiency.

[**Ans.** 2.3, 66.31%]

9.15 An ideal diesel cycle operates on a pressure of 1 bar and a temperature of 27°C at the beginning of compression and a pressure of 2 bar at the end of adiabatic expansion. Calculate the amount of heat required to be supplied per kg of air if the ideal thermal efficiency is taken as 60%. Take c_v = 0.717 kJ/kg.K.

[**Ans.** 537.75 kJ/kg]

9.16 The pressure and temperature of a diesel cycle at the start are 1 bar and 17°C. The pressure at the end of expansion is 2 bar. Find the air-standard efficiency. Assume $\gamma = 1.4$

[**Ans.** 61.14%]

9.17 A diesel cycle operates at pressure of 1 bar at the beginning of compression

and the volume is compressed to 1/15 of the initial volume. Heat is then supplied until the volume is twice that of the clearance volume. Determine the mean effective pressure. Take $\gamma = 1.4$

[**Ans.** 6.69 bar]

9.18 A semi diesel engine works on dual combustion cycle. The pressure and temperature at the beginning of the compression is 1 bar and 27°C respectively and the compressor ratio being 12. If the maximum pressure is 50 bar and heat received at constant pressure is for 1/30th of the stroke, find the work done per kg of air and the thermal efficiency. Take $c_p = 1.004$ kJ/kg.K, $c_v = 0.717$ kJ/kg.K.

[**Ans.** 537 kJ/kg, 69.27%]

9.19 A compression-ignition engine has a compression ratio of 10 and 2/3 of heat of combustion is liberated at a constant volume and the remainder at constant pressure. The pressure and temperature at the beginning are 1 bar and 27°C and the maximum pressure is 40 bar. Find the temperature at the end of compression and expansion, if it follows the law $pV^{1.35}$ = constant, and $\gamma = 1.4$.

[**Ans.** 398.6°C, 347.3°C]

9.20 An Otto cycle engine having a clearance volume of 250 cc has a compression ratio of 8. The ratio of pressure at constant volume is 9. If the initial pressure is 1 bar, find the work done per cycle and the theoretical mean

effective pressure. Take $\gamma = 1.4$

[**Ans.** 1946.1J/cycle, 11.12 bar]

9.21 Find the m.e.p for the ideal air-standard Otto cycle having a maximum pressure of 40 bar and minimum pressure of 1 bar. The compression ratio is 5:1. Take $\gamma = 1.4$

[**Ans.** 9.04 bar]

9.22 An engine working on Otto cycle, the ratio of temperature at the beginning of compression is 300 K. If the ideal air-standard efficiency = 0.5, calculate the compression ratio of the engine. If the peak temperature of the cycle is 1150K, calculate the temperature when the piston is at BDC during expansion stroke.

[**Ans.** 9.66, 575 K]

9.23 A petrol engine working on Otto cycle has a maximum pressure of 50 bar. If the pressure during combustion is 12.286, find the compression ratio and also the ratio of peak temperature to inlet temperature. Take p_1= 1 bar and t_1= 27°C.

[**Ans.** 6 8.34]

9.24 An Otto cycle takes in air at 300 K. The ratio of maximum to minimum temperature is 6. Find out the optimum compression ratio for the maximum work output of the cycle.

[**Ans.** 9 391]

9.25 The pressure and temperature of a diesel cycle at the start are 1 bar and 20°C respectively and the compression

ratio is 19. The pressure at the end of expansion is 2.5 bar. Find the percentage of working stroke at which heat is supplied and heal supplied per kg of air. Assume $\gamma = 1.4$ and $c_p = 1.004$ kJ/kg.K.

[**Ans.** (i) 7.11% (u) 781.46 kJ/kg]

9.26 An oil engine works on diesel cycle, the compression ratio being 19. The temperature at the start of compression is 17°C and 700 kJ of heat is supplied at constant pressure per Kg of air and attains a temperature of 417°C at the end of adiabatic expansion. Find the air-standard efficiency of the cycle. What would be the theoretical work done per kg of air. Take $c_v = 0.717$ kJ/kg K and $\gamma = 1.4$.

[**Ans.** 61.32%, 419.20 kJ]

9.27 An internal combustion engine works on Diesel cycle with a compression ratio of 8 and expansion ratio of 9. Calculate the air-standard efficiency Assume $\gamma = 1.4$

[**Ans.** 51.76%]

9.28 A Diesel engine works on Diesel cycle with a compression ratio of 16 and cut-off ratio of 1.8. Calculate the thermal efficiency assuming $\gamma = 1.4$

[**Ans.** 62.38%]

9.29 A compression-ignition engine works on dual combustion cycle. The pressure and temperature at the beginning of compression are 1 bar and 27°C respectively and the pressure at the end of compression is 25 bar. If 420

kJ of heat is supplied per kg of air during constant volume heating and the pressure at the end of adiabatic expansion is found to be 3 bar, find ihe ideal thermal efficiency. Assume c_p= 1.004 J/kg.K and c_v = 0 717 k.J/kg K.

[**Ans.** 58.27%]

9.30 The cycle of an internal combustion engine with isochoric heat supply is performed with the compression ratio equal to 8. Find heat supplied to the cycle and the useful work, if the removed heat is 500 kJ/kg and the working fluid is air.

[**Ans.** 1148.63 kJ/kg, 648.63 kJ/kg]

9.31 The initial parameters (at the beginning of compression) of the cycle of an internal combustion engine with

isobaric heat supply are 0.1 MPa and 80°C. The compression ratio is 16 and the heat supplied is 850 kJ/kg. Calculate the parameters at the characteristic points of the cycle and the thermal efficiency, if the working fluid is air.

[**Ans.** $p_2 = 48.5$ bar, $p_3 = 48.5$ bar, $p_3 = 2.26$ bar, $T_1 = 1070.1$ K, $T_3 = 1916$ K, $T_4 = 797.73$, $\eta_{th} = 62.5\%$]

9.32 The pressure ratio $\alpha_p = 1.5$ in the process of isochoric heat supply for the cycle at an internal combustion engine with a mixed supply of heat = 1034 kJ/kg and the compression ratio = 19. Find the thermal efficiency and temperature at the characteristic points of the cycle if the initial parameters are 0.09 MPa and 70°C and the working substance is air.

[**Ans.** $\eta_{th} = 62\%$, $T_2 = 956.9$ K, $T_3 = 1439.5$ K, $T_4 = 2122.86$ K, $T_5 = 890$ K]

9.33 The parameters of the initial state of one kilogram of an in the cycle of an internal combustion engine are 0.095 MPa and 65°C. The compression ratio is 11. Compare the values of the thermal efficiency for isobaric and isochoric heat supply in amounts of 800 kJ, assuming that $\gamma = 1.4$.

[**Ans.** η_{tp} = 59.7 %, htv = 61.7%]

9.34 Find the thermal efficiency of the cycle of an internal combustion engine with a mixed heat supply, if the minimum temperature of the cycle is 85°C and the maximum temperature is 1700 K. The compression ratio is 15 and the pressure ratio in the process of heat supply is 1.4. The working fluid is air.

[**Ans.** η_{th} = 69.25%]

9.35 A reversed Carnot cycle uses 0.1 kg of ammonia as a working fluid. At the beginning of the isothermal expansion, the ammonia is at 80 kPa, 30 percent quality; at the end of the isothermal expansion, the ammonia is dry and saturated. Heat is rejected at 20°C. Determine the change in internal energy of the ammonia during the isothermal expansion and the work input per cycle.

[**Ans.** 89.0 kJ, 23.9 kJ]

9.36 A closed-system Stirling cycle using helium as the working fluid operates with maximum pressure and temperature of 1200 kPa, 1000°C. The cycle has an overall volume ratio of 4.2 and minimum pressure and temperature

of 75 kPa, 61°C. Determine the thermal efficiency of the cycle.

9.37 For an Ericsson cycle as shown in Fig. 5.11, p_1 = 500 kPa, T_1 = 600°C, p_3 = 100 kPa, T_3 = 60°C, m = 0.220 kg/s, and the working fluid is argon. Determine the thermal efficiency.

9.38 A closed-system Stirling cycle using helium as working fluid has an overall volume ratio 3.8 and an overall temperature ratio of 3.0. Maximum pressure and temperature in the cycle are 2 MPa and 590°C. Determine the power output of the cycle for a heat input rate of 16 kW.

9.39 The overall temperature and volume ratios of a closed-system

Ericsson cycle are 3.0 and 4.8, respectively. The maximum temperature in the cycle is 640°C and maximum pressure of 3.5 MPa. Determine the cycle thermal efficiency.

9.40 In a Stirling cycle the volume varies between 0.03 and 0.06 m^3, the maximum pressure is 0.2 MPa, and the temperature varies between 540°C and 270°C.The working fluid is air

1. Find the work done per cycle and the efficiency for the simple cycle
2. Find the efficiency for the cycle with an ideal regenerator

[**Ans.** (a) 53.7 kJ/kg, 27.7%, (b) 33.2%]

9.41 An Ericsson cycle operating with an ideal regenerator works between 1100 K and 288 K. The pressure at the beginning of isothermal compression is 1.013 bar. Determine the compressor

and turbine work per kg of air, and the cycle efficiency.

[**Ans.** 121.8 kJ/kg, 465 kJ/kg, 0.738]

9.42 A Stirling regenerative air engine works between temperature of 400°C and 15°C. The ratio of isothermal expansion is 3. Calculate the ideal efficiency with and without the regenerator of efficiency 80%.

[**Ans.** 57.3%, 45.4%]

ANSWERS TO MULTIPLE-CHOICE QUESTIONS

1. c
2. b
3. d
4. b
5. a
6. a
7. d
8. c
9. d
10. a
11. b
12. b
13. d
14. c
15. b

16. a
17. b
18. c
19. c
20. a
21. a
22. d
23. a
24. c
25. d
26. c
27. a
28. b
29. b
30. b
31. c
32. d
33. c
34. d
35. d
36. b
37. c
38. a
39. a
40. a
41. c
42. c
43. a
44. c
45. b
46. a
47. a and c
48. d
49. a
50. d
51. a
52. a

Chapter 10

Internal Combustion Engine Systems

10.1 ❒ INTRODUCTION

An engine may be defined as a device which converts one form of energy into mechanical energy. Mechanical energy can be further easily converted into electrical energy which is easier to transport and control. Heat engine is a cyclic device which transforms heat energy into mechanical energy.

Heat engines are classified into the following:

1. External combustion engines
2. Internal combustion engines

In an external combustion engine, the fuel is burned outside the engine and the generated heat is supplied to the working fluid of the engine for power generation. Steam engine is an example of an external combustion engine in which fuel is burned in the boiler to generate steam, which is used in the engine for power generation. The working fluid is not mixed with fuel, and the same working fluid (water in the form of steam) is repeatedly used in the system.

In an internal combustion (IC) engine, the fuel is mixed with air and burned inside the engine to generate power. In this case, the same working fluid (air-fuel mixture) cannot be used again in

the cycle. Some examples of internal combustion engine are spark ignition (SI) engine, compression ignition (CI) engine, etc.

IC engines offer the following advantages over external combustion engines:

1. Greater mechanical simplicity
2. Lower weight to output ratio
3. Higher overall thermal efficiency
4. Less water requirement
5. Easy and quick starting

10.2 ❐ CLASSIFICATION OF INTERNAL COMBUSTION ENGINES

Figure 10.1 shows the classification of IC engines on the basis of cycle of operation into cylinder, type of fuel, method of supply of fuel, type of ignition, among others.

Figure 10.1 *IC engine classification*

1. **Basic engine design:** Reciprocating engines, rotary (Wankel) engines
2. **Working cycle:** Otto cycle and diesel cycle
3. **Number of strokes:** Four-stroke and two-stroke engines
4. **Fuel:** Gasoline (or petrol), compressed natural gas (CNG), liquefied petroleum gas (LPG), diesel oil (light diesel oil—LDO and high speed diesel oil—HSD), fuel oil, alcohols (methanol, ethanol)

Figure 10.2 *Classification of IC engines as per cylinder arrangement: (a) Inline engine, (b) Vee engine, (c) Radial engine, (d) Opposed piston engine, (e) Opposed cylinder engine, (f) Delta type engine*

5. **Fuel supply and mixture preparation:** Carburetted type and injection type
6. **Method of ignition:** CI (compression-ignition), SI (spark-ignition) engines: battery ignition or magneto ignition
7. **Method of cooling:** Water cooled or air-cooled
8. **Cylinder arrangement (Fig. 10.2):** Inline, V, or Vee, radial, opposed piston and opposed cylinder, delta
9. **Valve or port design and location:** Overhead valves (I-head), side valve (L-head); in two-stroke engines: cross scavenging, loop scavenging, uniform scavenging
10. **Application:** Automotive engines for land transport, marine engines for propulsion of ships, aircraft engines for aircraft propulsion, industrial engines, prime movers for electrical generators

10.3 ❐ CONSTRUCTION FEATURES

1. **Four-stroke spark-ignition engine:** The cross section of a four-stroke spark-ignition engine is shown in Fig. 10.3. The major components of the engine are as follows:
    1. **Cylinder block:** It is the main supporting structure for the various components. The cylinder head is mounted on the cylinder block. The bottom of the cylinder block is called the crank case. The lubricating oil is kept in the crank case sump.
    2. **Cylinder:** It is a cylindrical vessel in which the piston reciprocates.

Figure 10.3 *Cross-section of a spark-ignition engine*

    3. **Piston:** It is a cylindrical component fitted into the

    cylinder forming the moving boundary of the combustion system.
    4. **Combustion chamber:** It is the space between the cylinder and the piston top where combustion takes place.
    5. **Inlet manifold:** It is the pipe through which air or air-fuel mixture is drawn in the cylinder.
    6. **Exhaust manifold:** It is the pipe through which the products of combustion escape into the atmosphere.
    7. **Inlet and exhaust valves:** The valves are used for regulating the incoming charge into the cylinder (inlet valve) or discharging the products of combustion (exhaust valve) from the cylinder.
    8. **Spark plug:** It is a component to initiate the combustion process in SI engine.
    9. **Carburettor:** It is used for mixing fuel and air in correct proportion in SI engine.
    10. **Connecting rod:** It is used to interconnect the piston and the crank to transmit force from the piston to the crankshaft.
    11. **Crankshaft:** It is used to convert reciprocating motion of the piston into rotary motion of the output shaft.
    12. **Piston rings:** They provide a tight seal between the piston and the cylinder wall.
    13. **Gudgeon pin:** It connects the small end of the connecting rod to the piston.
    14. **Camshaft:** It is a shaft on which cams are mounted to operate the valves. It is driven from the cam shaft by gears.
    15. **Flywheel:** It absorbs surplus energy during working stroke and delivers during idle stroke.
2. **Four-stroke compression-ignition engine:** Except for the spark plug and carburettor, all other components are the same as for the SI engine. It also requires a fuel pump and fuel nozzle. Figure 10.4 shows outline of a CI engine.

Figure 10.4 *Outline of diesel engine*

3. **Two-stroke SI engine:** In the case of two-stroke engine, the valves are replaced by the exhaust port and the transfer port. The charging of the cylinder with the air-fuel mixture takes place through the carburetor and ignition by a spark plug. Other components are same as for a four-stroke engine.

4. **Two-stroke CI engine:** Here, the carburettor and the spark plug are replaced by a fuel pump and fuel nozzle. All other components are same as for two-stroke SI engine.

10.4 ❐ WORKING OF IC ENGINES

10.4.1 Four-stroke Spark-ignition Engine

The details of various processes of a four-stroke spark-ignition engine with overhead valves are shown in Fig. 10.5. Within the four strokes, there are five events to be completed namely, suction, compression, combustion, expansion, and exhaust.

Figure 10.5 *Working principle of a four-stroke SI engine: (a) Suction stroke, (b) Compression stroke, (c) Expansion or power stroke, (d) Exhaust stroke*

1. **Suction stroke:** It starts when the piston is at the top dead centre (TDC) and is about to move downwards. The inlet valve is open and the exhaust valve is closed at this time. The charge consisting of the fuel-air-mixture is drawn into the cylinder. When the piston reaches the bottom dead centre (BDC), the suction stroke ends and the inlet valve closes.
2. **Compression stroke:** The charge is compressed to the clearance volume by the return stroke of the piston with both inlet and exhaust valves are closed. At the end of the compression stroke, the mixture is ignited with the help of spark plug to convert chemical energy of fuel to heat energy.
3. **Expansion stroke:** The high pressure of burnt gases forces the

piston towards BDC with both the valves in closed position to produce power.

4. **Exhaust stroke:** At the end of the expansion stroke, the exhaust valve opens with the inlet valve closed. The piston starts moving towards the TDC and sweeps the burnt gases out from the cylinder. The exhaust valve closes when piston has reached TDC.

10.4.2 Four-stroke Compression-ignition Engine

The four cycles of operation of a CI engine are shown in Fig. 10.6.

1. **Suction stroke:** Only air is induced during the suction stroke with the inlet valve open and the exhaust valve closed.

Figure 10.6 *Cycle of operation of a four-stroke CI engine: (a) Suction, (b) Compression, (c) Expansion, (d) Exhaust*

2. **Compression stroke:** The sucked air is compressed into the clearance volume with both valves closed.
3. **Expansion stroke:** Fuel injection starts nearly at the end of the compression stroke, resulting in combustion. The products of combustion expand and both the valves remain closed.
4. **Exhaust stroke:** The piston moves from BDC to TDC and pushes out the products of combustion with the exhaust valve open and intake valve closed.

10.4.3 Two-stroke Spark-ignition Engine

In a two-stroke engine, the cycle is completed in one revolution of the crankshaft. The filling process is

accomplished by the charge compressed in the crankcase or by a blower. The induction of the compressed charge moves out the products of combustion through the exhaust ports. Therefore, two piston strokes are required for these two operations. Two strokes—one for compressing the fresh charge and the other for expansion or power stroke—are sufficient to complete the cycle. Figure 10.7 shows the crankcase scavenged two-stroke engine.

10.4.4 Two-stroke Compression-ignition Engine

Figure 10.8 shows a two-stroke compression-ignition engine. It is a crankcase scavenged type engine. S is the plate valve for admission of air in the crank case, E is the exhaust port, and

A is the port in the cylinder communicating to the crankcase through the cylinder block casting; cams and valves are not required. The four operations—air induction, air compression and fuel injection, expansion, and exhaust are completed in two strokes, that is, in one revolution of crankshaft.

In the upward motion of the piston, suction is created in the crankcase and air enters through the plate valves for full 180° of crank rotation as shown in Fig. 10.8(a). Above the piston, the compression starts after both the points have been covered by the piston. At the end of compression, fuel is injected and ignites at TDC giving products of

combustion at high pressure.

On downward stroke, the high pressure products of combustion expand and the air below the piston compresses, closing the plate valves, as shown in Fig. 10.8(b). As soon as the piston uncovers the exhaust port, the products of combustion are released into the atmosphere. A little later, on the downward stroke, the other port communicating with the crankcase gets uncovered and the air compressed in the crankcase gets transferred to the space above the piston as shown in Fig. 10.8(c).

Figure 10.7 *Crankcase scavenged two-stroke SI engine*

Figure 10.8 *Working of two-stroke C.I. Engine*

10.5 ❒ COMPARISON OF FOUR-STROKE AND TWO-STROKE ENGINES

Table 10.1 shows a comparison of four-stroke and two-stroke engines.

Table 10.1 *Comparison of two-stroke and four-stroke engines*

10.6 ❐ COMPARISON OF SI AND CI ENGINES

Table 10.2 gives the comparison of SI and CI engines.

Table 10.2 *Comparison of SI and CI engines*

10.7 ❐ MERITS AND DEMERITS OF TWO-STROKE ENGINES OVER FOUR-STROKE ENGINES

10.7.1 Merits

1. A two-stroke engine gives twice as many power strokes as a four-stroke cycle engine at the same engine speed; therefore, a two-stroke engine of the same size should develop twice the power of a four-stroke engine. In practice, the actual power

developed by a two-stroke engine is about 1.7 to 1.8 times the power developed by a four-stroke engine of the same dimensions and speed. This happens because some of the power is used for compressing the charge in the crank case and the effective stroke is less than the actual stroke.

2. For the same power developed, the two-stroke engine is much lighter, less bulky, and occupies less floor area. Therefore, it is more suitable for uses in marine engines and transport purposes.
3. It provides mechanical simplicity as valves, rocker arms, push-rods, cam, and cam shafts are not required. The friction loss is also less, and therefore, it gives higher mechanical efficiency.
4. The two-stroke engines are much easier to start.
5. A crankcase compression and valve-less type two-stroke engine can run in either direction, which is useful in marine applications.
6. The initial cost of the engine is considerable less.
7. The weight/kW ratio is considerably less.

10.7.2 Demerits

1. The thermodynamic efficiency of an engine is only dependent on the compression ratio. The effective compression ratio for a two-stroke engine is less than that for four-stroke engine for the same stroke (actual) and clearance volume. Therefore, the thermodynamic efficiency of two-stroke cycle is always less than a four-stroke cycle engine.
2. The actual efficiency of a two-stroke cycle is less than a four-stroke cycle engine because greater overlapping of ports is necessary in a two-stroke engine for effective scavenging. A portion of fresh charge in the case of an SI engine always escapes unused through the exhaust ports; therefore, the specific fuel consumption is usually higher.
3. As the power-strokes per minute are twice the power stroke of four-stroke cycle engines, the capacity of the cooling system used must be higher. The cooling of the engine also presents difficulty as the quantity of heat removed per minute is large. Due to firing in each revolution, the piston is likely to get overheated and oil cooling of the piston is necessary.
4. The consumption of lubricating oil is sufficiently large because of high operating temperatures.
5. Sudden release of the gases makes the exhaust noisier.

6. A two-stroke petrol engine with crankcase compression 50–60% of the swept volume is filled with fresh charge, whereas a four-stroke petrol engine contains 80–95%. This happens because the space occupied by the rotating parts in the crankcase prevents a full charge being sucked in.
7. The scavenging is not complete, particularly in high speed engines, as very short time is available for exhaust; hence, the fresh charge is highly polluted. This can be reduced using an opposed piston two-stroke diesel engine which provides unidirectional scavenging.
8. The turning moment of a two-stroke engine is more non-uniform as against with four-stroke engine, so it requires heavier flywheel and strong foundation.

10.8 ❒ VALVE TIMING DIAGRAMS

10.8.1 Four-stroke SI Engine

The valve timing diagram shows the regulation of the positions in the cycle at which the valves are set to open and close. The valve timing diagrams for low and high speed four-stroke S.I. engine are shown in Fig. 10.9. Typical valve timings are given in Table 10.3.

10.8.2 Four-stroke CI Engine

A typical valve timing diagram for a

four-stroke CI engine is shown in Fig. 10.10. The typical timing valves are also displayed.

IVO up to 30° before TDC; IVC up to 40° after BDC

EVO about 45° before BDC; EVC about 30° after TDC

Fuel valve opens (FVO) about 15° before TDC; FVC about 25° after TDC.

10.8.3 Two-stroke SI Engine

The transfer port opens 35° before the BDC and closes 35° after the BDC. The exhaust port opens 45° before the BDC and closes 45° after the BDC. The spark ignition occurs 20° before the TDC. The typical valve timing diagram for two-

stroke engine is shown in Fig. 10.11.

Figure 10.9 *Valve timing for low and high speed four-stroke SI engine: (a) Low speed engines, (b) High speed engines*

Table 10.3 *Typical valve timings for four-stroke SI engines*

Figure 10.10 *Valve timing diagram four-stroke CI engine*

Figure 10.11 *Typical valve timing diagram of a two-stroke SI engine*

10.8.4 Two-stroke CI Engine

The transfer port opens 45° before the BDC and closes 45° after the BDC. The exhaust port opens 60° before the BDC and closes 60° after the BDC. The fuel valve opens 15° before the TDC and closes 20° after the TDC. The typical valve timing diagram is shown in Fig. 10.12.

Figure 10.12 *Valve timing diagram for a two-stroke CI engine*

10.9 ❐ SCAVENGING PROCESS

At the end of the expansion stroke, the combustion chamber of a two-stroke engine is left full of products of combustion as there is no exhaust stroke available to clear the cylinder of burnt gases. The process of clearing the cylinder, after the expansion stroke, is called scavenging process. The scavenging process is the replacement of the products of combustion in the cylinder from the previous power stroke with fresh-air charge to be burned in the next cycle.

There are three types of scavenging systems as follows:

1. **Uniflow scavenging system:** In this system, as shown in Figs 10.13(a) and (b), air enters the cylinder from one end and leaves from the other end. Air acts like an ideal piston and pushes out the residual gas in the cylinder and replaces it with

fresh charge. Due to absence of any eddies or turbulence, in a uni-flow scavenging system, it is easier to push the products of combustion out of the cylinder without mixing with it and short circuiting. Thus, this system has the highest scavenging efficiency.

2. **Cross scavenging:** This process shown in Fig. 10.14. It employs inlet and exhaust ports placed in the opposite sides of the cylinder wall. The air moves up to combustion chamber on one side of the cylinder and then down on the other side to flow out of the exhaust ports. This process requires that air should be guided by the use of either a suitably shaped detector formed on piston top or by the use of inclined ports. The main disadvantage of this system is that the scavenging air is not able to get rid of the layer of exhaust gas near the wall, resulting in poor scavenging. Some of the fresh charge also goes directly in the exhaust port.

Figure 10.13 *Uni-flow scavenging system: (a) Exhaust valve, (b) Opposed piston*

Figure 10.14 *Cross scavenging*

Figure 10.15 *Loop or reverse scavenging*

3. **Loop or Reverse Scavenging:** Figures 10.15(a) and (b) show the loop or reverse scavenging system. This avoids the short-circuiting of the cross-scavenged engine and improves upon the scavenging efficiency. The inlet and exhaust ports are placed on the same side of the cylinder wall. The major mechanical problem with this system is that of obtaining an adequate oil supply to the cylinder wall consistent with reasonable lubricating oil consumption and cylinder wear.

10.10 ❒ APPLICATIONS OF IC ENGINES

The detail applications with capacity and type of engine used are listed in Table 10.4.

Table 10.4 *Applications of IC engines*

10.11 ❐ THEORETICAL AND ACTUAL *p-v* DIAGRAMS

10.11.1 Four-stroke Petrol Engine

The theoretical and actual *p-v* diagrams for a four-stroke petrol engine are shown in Figs. 10.16(a) and (b), respectively.

The theoretical *p-v* diagram is drawn with the following assumptions:

1. Suction and exhaust take place at atmospheric pressure through 180° rotation of crank.
2. Compression and expansion take place through 180° rotation of crank.
3. Compression and expansion processes are isentropic.
4. The combustion takes place instantaneously at constant volume at the end of compression stroke.
5. Pressure suddenly falls to the atmospheric pressure at the end of expansion stroke.

The various processes of theoretical *p-v* diagram are as follows:

5-1: Suction stroke (p_a = const)

1-2: Compression stroke (pv^γ = const)

Figure 10.16 *Theoretical and actual p-v diagrams for a four-stroke petrol engine: (a) Theoretical, (b) Actual*

2-3: Instantaneous combustion (v = const)

3-4: Expansion stroke (pv^γ = const)

4-1: Sudden fall in pressure (v = const)

1-5: Exhaust stroke (p_a = const)

In practice, the actual conditions differ from the ideal as follows:

1. The suction of mixture in the cylinder is possible only if the

pressure inside the cylinder is below atmospheric pressure.

2. The burnt gases can be pushed out into the atmosphere only if the pressure of the exhaust gases is above atmospheric pressure.
3. The compression and expansion do not follow the isentropic law, as there will be heat exchange during these processes.
4. Sudden pressure rise is not possible after the ignition as combustion takes some time for completion and actual pressure rise is less than theoretical considered. The pressure increase takes place through some crank rotation, or increase in volume.
5. Sudden pressure release after the opening of expansion valve is not possible and it also takes place through some crank rotation.

If all these modifications are taken into account, then the cycle can be represented on *p*-*v* diagrams as shown in Fig. 10.16(b).

The area 4′-5-1-4′ representing negative work is called negative loop or pumping loop. This work is required for admitting the fresh charge and for exhausting the burnt gases. This loss of work is known as *pumping loss* and power consumed for this is known as

pumping power.

The net work per cycle of the engine is given by the area $(A_1 - A_2)$. This area $(A_1 - A_2)$ is always less than the area A as shown in Fig. 10.16(a) due to the actual deviations of operations from the theoretical ones.

If both the areas are represented in the form of rectangles taking v_s as base, then the ordinates give the mean effective pressures as shown in Fig. 10.17.

p_{ma} (Actual mean effective pressure) = p_{mt} (theoretical mean effective pressure) $\times$ DF (diagram factor)

10.11.2 Four-stroke Diesel Engine

The theoretical and actual *p-v* diagrams for a four-stroke diesel engine are shown in Figs. 10.18(a) and (b), respectively.

Figure 10.17 *p-v diagrams*

Figure 10.18 *Theoretical and actual p-v diagrams for a four-stroke diesel engine: (a) Theoretical, (b) Actual*

The theoretical *p-v* diagram for a four-stroke diesel engine is drawn with the following assumptions:

1. Suction and exhaust take place at atmospheric pressure through 180° of crank rotation.
2. Compression and expansion take place during 180° of crank rotation.
3. Compression and expansion are isentropic.
4. The combustion takes place at constant pressure during a small part of expansion stroke.
5. Pressure suddenly falls to atmospheric pressure at the end of expansion stroke.

With the above assumptions, the working cycle can be represented on a *p-v* diagram as shown in Fig. 10.18(a),

and it is similar to the theoretical diesel cycle. The various processes of theoretical p-v diagram are as follows:

5-1: Suction stroke (p_a = const)

1-2: Compression stroke (pv^γ = const)

2-3: Constant pressure combustion (p = const)

3-4: Expansion stroke (pv^γ = const)

4-1: Sudden fall in pressure (v = const)

1-5: Exhaust stroke (p_a = const)

However, in practice, the actual conditions differ from the ideal described as follows:

1. The suction of the air inside the cylinder is possible only if the pressure inside the cylinder is below atmospheric.
2. Exhausting of gasses is possible only if the pressure of the exhaust gases is above atmospheric pressure.
3. The compression and expansion do not follow the isentropic process, as there are heat and pressure losses.
4. The combustion at constant pressure is not possible as the fuel will not burn as it is introduced into the cylinder.
5. The sudden pressure release after the opening of expansion valve is not possible and it takes place through some crank rotation.

The operations of the cycle, taking the modifications into account, are represented on the *p*-*v* diagram in Fig. 10.18(b). Actual area $(A_1 - A_2)$ on *p*-*v* diagram per cycle is less than theoretical.

10.11.3 Two-stroke Petrol Engine

The theoretical and actual *p*-*v* diagrams for a two-stroke petrol engine are shown in Fig. 10.19. The following assumptions are made for drawing the theoretical *p*-*v* diagram:

1. The expansion during power stroke and compression during compression stroke are isentropic.
2. The combustion takes place instantaneously at constant volume at the end of compression.
3. The pressure falls instantaneously to the atmospheric pressure as the piston uncovers the exhaust ports during power stroke.
4. The scavenging takes place at atmospheric pressure.

It may be observed from Fig. 10.19(a) that the compression of the charge starts from '1' instead of point 6.

Effective compression ratio,

where v_{se} = effective stroke volume =

v_{sa} = actual stroke volume =

Figure 10.19 *Theoretical and actual p-v diagrams for a two-stroke petrol engine: (a) Theoretical, (b) Actual*

L_e, L_a = effective and actual stroke lengths, respectively

d = cylinder diameter

The various processes are as follows:

1-2: isentropic compression ($pv^\gamma = c$)

2-3: instantaneous combustion ($v = c$)

3-4: isentropic expansion ($pv^\gamma = c$)

4-1: release of burned charge to atmosphere ($v = c$)

1-6 and 6-1: sweeping out of exhaust gases to atmosphere

Point 5: inlet port opens

5-6 and 6-5: charging the cylinder with fresh charge and scavenging action. Actual theoretical mean effective pressure,

p_{ma}, p_{me} = actual mep on the basis of actual and effective stroke respectively. In practice, the actual conditions differ from the ideal as described below:

1. The compression and expansion processes do not follow the isentropic law strictly.
2. Instantaneous combustion at the end of compression is not possible. The actual pressure rise takes place through some crank angle, resulting in rounding of the diagram.
3. The scavenging always takes place above atmospheric pressure.
4. Instantaneous fall of pressure at the time of release is not possible.

10.11.4 Two-stroke Diesel Engine

The theoretical and actual *p*-*v* diagrams for a two-stroke diesel engine are shown in Fig. 10.20. The various processes are as follows:

1-2: isentropic compression ($pv_y = c$)

2-3: instantaneous combustion ($p = c$)

3-4: isentropic expansion ($pv^{\gamma} = c$)

4-1: release of burned charge to atmosphere ($v = c$)

1-6 and 6-1: sweeping out of exhaust gases to atmosphere

5-6 and 6-5: charging the cylinder with fresh charge and scavenging action.

Point 5: inlet port opens.

Figure 10.20 *Theoretical and actual p-v diagrams for a two-stroke diesel engine: (a) Theoretical, (b) Actual*

10.12 ❐ CARBURETION

In SI engines, the air and fuel are mixed outside the engine cylinder and partly evaporated mixture is supplied to the engine. The process of preparing this mixture is called *carburetion***.** The

device used for this purpose is known as *carburettor*. The carburettor atomises the fuel and mixes it with air. This complicated process is achieved in the induction system, which is shown in Fig. 10.21. The pipe that carries the prepared mixture to the engine cylinder is called the intake manifold.

During the suction stroke, vacuum is created in the cylinder which causes the air to flow through the carburettor and the fuel to be sprayed from the fuel jets. Due to high volatility of the SI engine fuels, most of the fuel vaporises and forms a combustible fuel-air mixture. However, some of the larger droplets may reach the cylinder in the liquid form and must be vapourised and mixed

with air during the compression stroke before ignition takes place by the electric spark.

Figure 10.21 *Induction system for SI engine*

Table 10.5 *F:A ratios for various running conditions of SI engines*

The fuel-air ratios for various running conditions of SI engines are given in Table 10.5.

10.12.1 Simple Carburettor

The details of a simple carburettor are shown in Fig. 10.22. It consists of a float chamber, fuel discharge nozzle and a metering orifice, a venturi, a throttle valve, and a choke. A float and a needle valve system maintains a constant level of gasoline in the float chamber. If the

amount of fuel in the float chamber falls below the designed level, the float goes down, thereby opening the fuel supply valve and admitting fuel. When the designed level is reached, the float closes the fuel supply valve, thus stopping additional fuel flow from the supply valve and stopping additional fuel flow from the supply system. The float chamber is vented either to the atmosphere or to the upstream side of the venturi.

During suction stroke, air is drawn through the venturi (or choke tube). As the air passes through the venturi, the velocity increases, reaching a maximum at the venturi throat and pressure decreases to a minimum. From the float

chamber, the fuel is fed to a discharge jet, the tip of which is located in the throat of the venturi, fuel is discharged into the air stream. To avoid overflow of fuel through the jet, the level of the liquid in the float chamber is maintained at a level slightly below the tip of the discharge jet.

The throttle valve controls the amount of charge delivered to the cylinder, which is situated after the venturi tube. As the throttle is closed, less air flows through the venturi tube and less quantity of the air-fuel mixture is delivered to the cylinder; hence, the power output is reduced. The reverse takes place when the throttle is opened.

10.12.2 Compensating Jet

The function of the compensating jet is to make the mixture leaner as the throttle opens progressively. As shown in Fig. 10.23, the compensating jet is connected to the compensating well which is vented to the atmosphere. The compensating well is supplied with fuel from the main float chamber through a restricting orifice. With the increase in air flow rate, there is a decrease in the fuel level in the compensating well, with the result that fuel supply through the compensating jet decreases. The compensating jet, thus, progressively makes the mixture leaner as the main jet progressively makes the mixture richer.

Figure 10.22 *Simple carburettor*

Figure 10.23 *A compensating jet device*

10.12.3 Theory of Simple Carburettor

The air from the atmosphere is sucked through the carburettor by the pressure difference across it created when the piston moves on its suction stroke. The velocity of the air increases as it passes through the venturi and reaches maximum at venturi throat (Fig. 10.24). The pressure also changes and is maximum at section 2-2, because this is the minimum area in the induction track. The fuel is sucked through the nozzle because of suction created in the venturi.

Figure 10.24 *Principles of a simple carburettor*

1. **Approximate analysis (neglecting compressibility of air):** Let z be the height in metre of fuel nozzle tip higher than the float chamber level. Assuming initial velocity of air to be negligible ($c_1 = 0$), density of air to be constant and considering sections at entrance and venturi throat, by applying, Bernoulli's theorem, we get

   where p_1, p_2 = pressure at sections 1 and 2, respectively, N/m^2

c_2 = velocity of air at section 2, m/s

ρ_a = density of air, kg/m^3

Mass of air per second,

where A_2 = area of venturi throat, m^2

Similarly, for the flow of fuel, we have

where ρ_f = constant density of fuel, kg/m^3

c_f = velocity of flow of fuel, m/s

Mass flow rate of fuel,

where A_f = cross-sectional area of fuel nozzle, m^2

Taking the coefficient of discharge of fuel nozzle and venturi into account, we have

where C_d = coefficient of discharge.

If $z = 0$.

2. **Exact analysis:** Considering compressibility of air and applying steady flow energy equation to sections 1 and 2, we get

where h_1, h_2 = enthalpies at sections 1 and 2 respectively

Since $Q = 0$, $W = 0$ and $c_1 = 0$

For isentropic flow between the atmosphere and venturi throat, we have

where v = specific volume

Since

Eq. (10.7) becomes

10.12.4 Limitations of Single Jet Carburettor

The drawbacks of single jet carburettor are as follows:

1. A single jet carburettor cannot provide a very rich mixture as required at the time of starting the engine. This is because at low speed (starting or idling), the pressure difference causing the fuel flow is very small as the throttle is nearly closed. It is not possible to discharge fuel to make the mixture considerably rich.
2. It cannot provide a very rich mixture required for sudden acceleration of the engine.
3. For gradually increasing pressure difference over the jet (at higher speed of the engine), the weight of the petrol discharged from a single jet increases at a greater rate than does the air

supply. Hence, a single jet carburettor gives a progressively richer mixture as the air speed increases when set to given a correct mixture at low air speeds.

4. It cannot reduce the quantity of air flow during starting as required in cold weather conditions.
5. The automatic control of air and fuel according to the required conditions is not possible.

The carburettor used with a variable speed engine must fulfil all the following requirements:

1. It must atomise the fuel and mix it homogeneously with the air.
2. It must be able to run the engine smoothly without hunting or fuel wastage.
3. It must provide rich mixture during starting and idling.
4. It must provide a constant air-fuel ratio during normal running of the engine which is the maximum period.
5. It must provide a rich mixture required for quick acceleration of the engine.
6. It must be able to start the engine even in very cold weather conditions (during snowfall).
7. All operations should be automatic.

To fulfil the above requirements, the following devices are introduced.

1. A starting or a pilot jet (to start engine)
2. Compensating devices (to provide constant A:F ratio during normal operation conditions)
3. An automatic control of choke valve (to start the engine in cold weather)

10.12.5 Different Devices Used to Meet the Requirements of an Ideal Carburettor

The main function of the carburettor is to vapourise the petrol in the current of air created by means of engine suction and supply the required quantity of air and petrol mixture in proper proportion in accordance with the load on the engine and its speed.

In the design of a carburettor, the following points should be considered:

1. It is seen that for a gradually increasing pressure difference over the fuel jet, the mass of petrol discharged from a single jet increases at a greater rate than does the air supply. Hence, a carburettor of this type will give a progressively rich mixture as air speed is increased. There must be some device to maintain A: F ratio constant over a wide range of engine speed.
2. To ensure rapid and complete combustion, it is necessary that the fuel should be finely divided and intimately mixed with air supply. This can be done by proper design of venturi and inlet manifold.
3. For starting and accelerating, a rich mixture must be supplied momentarily but the supply should come to the correct mixture strength automatically when the engine attains the desired speed.
4. There must be a provision to supply extra-rich mixture to start the engine in a very cold weather (where temperature falls to

zero or sub-zero temperature).
5. The float in the float chamber of a carburettor must maintain the fuel level constant irrespective of load or speed of the engine.

10.12.6 Complete Carburettor

In order to satisfy the demands of an engine under all conditions of operations, the following additional systems are added to the simple carburettor:

1. Main metering system
2. Idling system
3. Power enrichment by economiszer system
4. Acceleration pump systems
5. Choke.

Main Metering System

The main metering system of a carburettor is designed to supply a nearly constant basis fuel-air ratio over a wide range of speeds and loads. This mixture corresponds approximately to best economy at full throttle (A/F ratio

$\approx$ 15.6 or F/A ratio 0.064). Since a simple or elementary carburettor tends to enrich the mixture at higher speeds automatic compensating devices are incorporated in the main metering system to correct this tendency. These devices are as follows:

1. Use of a *compensating jet* that allows an increasing flow of air through a fuel passage as the mixture flow increases.
2. Use of *emulsion tube* for air bleeding. In this device, the emphasis is on air bleeding alone.
3. Use of a tapered metering pin that is moved in and out of the main or auxiliary fuel orifice either manually or by means of some automatic mechanism changing the quantity of fuel drawn into the air charge.
4. Back-suction control or pressure reduction in the float chamber.
5. Changing the position or jet in the venturi. The suction action is highest at the venturi throat, therefore by raising the venturi the nozzle relatively moves to points with smaller suction and the flow of fuel is decreased.
6. Use of an auxiliary air valve or port that automatically admits additional air as mixture flow increases.

The main devices are explained in detail.

1. **Compensating jet device:** This device is shown in Fig. 10.23. In this device, in addition to the main jet, a compensating jet is

provided which is in communication with a compensating well.

The compensating well is open to atmosphere and gets its fuel supply from the float chamber through a restricting orifice. As the air flow increases, the level of fuel in the well decreases, thus reducing the fuel supply through the compensating jet. The compensating jet thus tends towards leanness as the main jet tends towards richness, the sum of the two remaining constant as shown in Fig. 10.25. At even higher rates of air flow, when the compensating jet has been emptied, air is bled through the compensating jet to continue the leanness effect, and incidentally, to assist in fuel atomisation.

2. **Emulsion tube or air bleeding device**: In modern carburettors, the mixture correction is done only by air bleeding. In this arrangement, the main metering jet is fitted about 25 mm below the petrol level and it is called a submerged jet (see Fig. 10.26). The jet is situated at the bottom of a well, the sides of which have holes which are in communication with the atmosphere. Air is drawn through the holes in the well, the petrol is emulsified, and the pressure difference across the petrol column is not as great as that in the simple or elementary carburettor. Initially, the petrol in the well is at a level equal to that in the float chamber. On opening the throttle this petrol, being subject to the low throat pressure, is drawn into the air. This continues with decreasing mixture richness as the boles in the central tube are progressively uncovered. Normal flow takes place from the main jet.

Figure 10.25 *Variation of air-fuel ratio vs air flow with main and compensating jet*

Figure 10.26 *Correction in modern carburettors by air bleeding*

3. **Back-suction control or pressure reduction method:** A

common method of changing the air-fuel ratio in large carburettors is the back-suction control as shown in Fig. 10.27. In this arrangement, a relatively large vent line connects the carburettor entrance (say point 1) with the top of the float chamber. Another line, containing a very small orifice line, connects the top of the float chamber with the venturi throat (say point 2). A control valve is placed in the large vent line. When the valve is wide open, the vent line is unrestricted the pressure in the float chamber equal to p_1, and the pressure difference acting on the fuel orifice is $(p_1 - p_2)$. If the valve is closed, the float chamber communicates only with the venturi throat and the pressure on the fuel surface will be p_2. Then Δp_f will be zero, and no fuel will flow. By adjusting the control valve, any pressure between p_1 and p_2 may be obtained in the float chamber, thus changing the quantity of fuel discharged by the nozzle.

Figure 10.27 *Back-suction control or pressure reduction method*

4. **Auxiliary valve carburettor:** An auxiliary valve carburettor is illustrated in Fig. 10.28. With an increase of engine load, the vacuum at the venturi throat also increases. This causes the valve spring to lift the valve admitting additional air and the mixture is prevented from becoming over-rich.
5. **Auxiliary part carburettor:** An auxiliary port carburettor is illustrated in Fig. 10.29. By opening the butterfly valve, additional air is admitted and at the same time the depression at the venturi throat is reduced, decreasing the quantity of fuel drawn in. This method is used in aircraft carburettors for altitude compensation.

Figure 10.28 *An auxiliary valve carburettor*

Figure 10.29 *An auxiliary port carburettor*

Idling System

It has already been shown that at idling and low lead, the engine requires a rich mixture (about A/F 12:1). However, the

main metering system not only fails to enrich the mixture at low air flows but also supplies no fuel at all at idling. For this reason, a separate idling jet must be added to the basic carburettor. An example of idling jet is shown in Fig. 10.30. It consists of a small fuel line from the float chamber to a little on the engine side of the throttle. This line contains a fixed fuel orifice. When the throttle is practically closed, the full manifold suction operates on the outlet to this jet. In addition, the very high velocity past be throttle plate increases the suction locally. Fuel can, therefore, be lifted by the additional height up to the discharge point, but this occurs only at very low rates of air flow. As the throttle is opened, the main jet gradually

takes over while the idle jet becomes ineffective. The desired air-fuel ratio for the idling jet is regulated manually to idle adjust, which is a needle valve controlling the air bleed.

Power Enrichment of Economiser System

As the maximum power range of operation (75% to 100% load) is approached, some device must allow richer mixture (A/F about 13: 1, F/A 0.08) to be supplied despite the compensating jet. Such a device is the meter rod economiser shown in Fig. 10.31. The name economiser is rather misleading. It stems from the fact that such a device provides a rich uneconomical mixture at high load demand without interfering with

economical operation in the normal power range. The meter rod economiser shown in Fig. 10.31, simply provides a large orifice opening to the main jet as the throttle is opened beyond a certain point.

The rod may be tapered or stepped. Other examples provide for the opening of auxiliary jets through some linkage to the throttle movement or through a spring action when manifold vacuum is lost as the throttle is opened.

Acceleration Pump System

It has already been shown that when it is desired to accelerate the engine rapidly, a simple carburettor will not provide the required rich mixture. Rapid opening of the throttle will be immediately

followed by an increased airflow, but the inertia of the liquid fuel will cause at least a momentarily lean mixture just when richness is desired for power. To overcome this deficiency, an *acceleration pump* is provided; an example is shown in Fig. 10.32. The pump consists of a spring-loaded plunger. A linkage mechanism is provided so that when the throttle is rapidly opened the plunger moves into the cylinder and forces an additional jet of fuel into the venturi. The plunger is raised again against the spring force when the throttle is partly closed. Arrangement is provided so that when the throttle is opened slowly, the fuel in the pump cylinder is not forced into the venturi but leaks past plunger or some

holes into the float chamber.

Figure 10.30 *Idling jet*

Figure 10.31 *Meter rod economiser*

Instead of the mechanical linkage shown, some carburettors have a plunger held up by a manifold vacuum. Wherever that vacuum is reduced by the rapid opening of the throttle, a spring forces the plunger down in a pumping action identical to that of the pump illustrated.

Choke

During cold starting period, at low cranking speed and before the engine has warmed up, a mixture much richer than usual mixtures (almost five to 10 times more fuel) must be supplied because a large fraction of the fuel will

remain liquid even in the cylinder, and only the vapour fraction can provide a combustible mixture with the air. The most common means of obtaining this rich mixture is by the use of a *choke,* which is a butterfly-type valve placed between the entrance to the carburettor and the venturi throat as shown in Fig. 10.33. By partially closing the choke, a large pressure drop can be produced at the venturi throat that would normally result from the amount of air flowing through the venturi. This strong suction at the throat will draw large quantities of fuel from the main nozzle and supply a sufficiently rich mixture so that the ratio of evaporated fuel to air in the cylinders is within combustible limits. Choke valves are sometimes made with a

spring loaded so that high-pressure drops and excessive choking will not result after the engine has started and has attained a higher speed.

Figure 10.32 *Acceleration pump*

Figure 10.33 C*hoke valve with spring-loaded bypass*

Some manufacturers make the choke operate automatically by means of a thermostat such that when the engine is cold, the choke is closed by a bimetallic element. After starting and as the engine warms up, the bimetallic element gradually opens the choke to its fully open position.

An alternative to the choke is the provision of auxiliary fuel jets that are opened manually or automatically only as required.

Example 10.1

A single jet carburettor has to supply 5 kg of air per minute. The air is at a pressure of 1.013 bar and at a temperature of 27°C. Calculate the throat diameter of the choke for air flow velocity of 90 m/s. Take the velocity coefficient as 0.8. Assume the flow to be incompressible and isentropic.

[IES, 1992]

Solution

Given that $\dot{m}_a$ = 5 kg/min, p_1 = 1.013 bar, T_1 = 273 + 27 = 300 K, c_2 = 90 m/s, C_{da} = 0.8

Applying Bernoulli's theorem between inlet and throat, we have

Density of air at inlet, ρ_{a1} =

Density of air at throat, ρ_{a2} =

Mass flow rate at throat, $\dot{m}_a = \rho_{a2} A_2 c_2$

throat dia, d_2 = 0.0325 m or 32.5 mm

Example 10.2

A six-cylinder, four-stroke engine,

80 mm in diameter and 120 mm stroke length runs at 3600 rpm. The volumetric efficiency of the engine is 0.8. If the maximum head causing the flow is limited to 11.765 cm of mercury, find the throat diameter of the venturi required. Find the diameter of the nozzle orifice if the desired A: F ratio is 15:1. Take C_{da} = 0.9, C_{df} = 0.7, ρ_a = 1.3 kg/m^3 and ρ_f = 720 kg/m^3.

Solution

Given that $i = 6$, $d = 0.08$ m, $L =$ 0.120 m, $N = 3600$ rpm, $\eta_{vol.} = 0.8$, $h_{Hg} = 11.765$ cm

Volume flow rate of air,

Mass flow rate of air, $\dot{m}_a = \rho_a \dot{V}_a =$ $1.3 \times 0.0868 = 0.113$ kg/s

$$h_w = h_{Hg} \times \gamma_{Hg} = 11.765\ 13.6 = 160 \text{ of water}$$

Example 10.3

The throat diameter of a carburettor is 80 mm and the nozzle diameter is 5.5 mm.The nozzle lip is 6 mm. The pressure difference causing the flow is 0.1 bar. Find (a) the air-fuel (A:F) ratio supplied by the carburettor neglecting nozzle lip. (b) A:F ratio considering nozzle lip and (c) the minimum velocity of air required to start the fluid flow. Neglect air compressibility. Take $C_{da} = 0.85$,

$C_{df} = 0.7$, $\rho_a = 1.2$ kg/m^3 and ρ_f =750 kg/m^3.

Solution

Given that $d_a = 80$ mm, $d_f = 5.5$ mm, $z = 6$ mm, $\Delta p = 0.1$ bar, $C_{da} = 0.85$, $C_{df} = 0.7$, $\rho_a = 1.2$ kg/m^3, $\rho_f = 750$ kg/m^3

1.
2.
3.

Example 10.4

A 4-stroke petrol engine of 2 litres capacity is to develop maximum power at 4000 rpm. The volumetric efficiency at this speed is 0.75 and the air-fuel ratio is 14:1. The venturi

throat diameter is 28 mm. The coefficient of discharge of venturi is 0.85 and that for fuel jet is 0.65. Calculate (a) the diameter of the fuel jet and (b) the air velocity at the throat.

The specific gravity of petrol is 0.76. Atmospheric pressure and temperature are 1 bar and 17°C respectively.

[IAS, 2002]

Solution

Given: V_s = 2 litres or 2×10^{-3} m^3, N = 4000 rpm, n = 2, η_{vol} = 0.75, A/F = 14/1, d_2 = 28 mm, C_{da} = 0.85, C_{df} = 0.65, ρ_f = 760 kg/m3, p_1 = 1 bar, T_1 = 17 + 273 = 290 K

1.

or $d_f = 28 \times 0.32733 = 9.165$ mm

$\therefore$ Diameter of fuel jet = 9.165 mm

2. Volume of air supplied per cycle = stroke volume $\times\ \eta_{vol}$

$$V_1 = 2 \times 10^{-3} \times 0.75 = 1.5 \times 10^{-3}\ m^3$$

Mass of this air per cycle at suction condition,

$$= 1.8 \times 10^{-3}\ kg/s$$

Mass of air supplied,

$$\dot{m}_a = A_2\ C_2\ \rho_{a2} \quad (\rho_{a1} = \rho_{a2})$$

or $c_2 = 81.2$ m/s

Example 10.5

The venturi of a simple carburettor has a throat diameter of 20 mm and the fuel orifice has a diameter of 1.12 mm. The level of petrol surface in the float chamber is 6.0 mm below the throat venturi. Coefficient of discharge for venturi and fuel

orifice are 0.85 and 0.78 respectively. Specific gravity of petrol is 0.75. Calculate (a) the air-fuel ratio for a pressure drop of 0.08 bar, (b) petrol consumption in kg/hr and (c) the critical air velocity. The intake conditions are 1.0 bar and 17°C. For air $c_p = 1.005$ and $c_v = 0.718$ kJ/kg-K.

[IAS, 2003]

Solution

Given: $d_2 = 20$ mm, $d_f = 1.12$ mm, $z = 6$ m, $C_{da} = 0.85$, $C_{df} = 0.78$, $s_f = 0.75$, $\Delta p = 0.08$ bar, $p_1 = 1.0$ bar, $T_1 = 290$ K, $c_p = 1.005$ kJ/kg.K, $c_v = 0.718$ kJ/kg.K

1. Fuel-air ratio,
2. Petrol consumption,

3. Critical air velocity

Example 10.6

The air-fuel ratio of a mixture supplied to an engine by a carburettor is 15. The fuel consumption of the engine is 7.5 kg/h. The diameter of the venturi is 2.2 cm. Find the diameter of fuel nozzle if the lip of the nozzle is 4 mm. Take $\rho_f = 750$ kg/m^3, $C_{da} = 0.82$, $C_{df} = 0.7$, atmospheric pressure = 1.013 bar and temperature = 25°C. Neglect compressibility effect of air.

Solution

Given that A: F = 15:1, $\dot{m}_f = 7.5$ kg/h, $d_a = 2.2$ cm, $z = 4$ mm, $\rho_f = 750$

kg/m^3, $C_{da} = 0.82$, $C_{df} = 0.7$, $p =$ 1.013 bar, $T = 273 + 25 = 298$ K

Density of air,

Mass flow rate of air,

Example 10.7

A four-stroke petrol engine of 1710 cm^3 capacity is to be designed to develop maximum power at 5400 rpm, when the mixture A:F ratio supplied to the engine is 13:1. Two carburettors are to be fixed. The maximum velocity of air is limited to 107 m/s. Find the diameters of venturi and fuel nozzle jet, if η_{vol}

(engine) = 0.7, C_{da} = 0.85, C_{df} = 0.66, nozzle lip = 6 mm and ρ_f = 750 kg/m^3. Atmospheric pressure and temperature are 1.013 bar and 300 K. Take c_p = 1 kJ/kgK.

Solution

Given that V_e = 1710 cm^3, N = 5400 rpm, A:F = 13, c_a = 107 m/s, η_{vol} = 0.7, C_{da} = 0.85, C_{df} = 0.66, z = 6 mm, ρ_f = 750 kg/m^3, p = 1.013 bar, T = 300 K, c_p = 1 kJ/kgK

Volume of air supplied to engine at STP,

Air flow through each carburettor,

Mass flow rate of air,

Air velocity through venturi,

Assuming the flow to be isentropic,

Volume of air passing through each carburettor,

$$V_2' = A_a c_a C_{da}$$

or $d_a = 2$ cm

Mass flow rate of fuel per carburettor,

Example 10.8

A 4-stroke petrol engine has a swept volume of 2.0 litres and is running at 4000 rpm. The volumetric efficiency at this speed is 75% and the air-fuel ratio is 14:1. The venturi throat diameter of the carburettor fitted to the engine is 30 mm. Estimate the air velocity at the throat if the discharge coefficient for air is 0.9. The ambient conditions are, pressure = 1.0 bar, temperature = 20°C. Calculate the diameter of the fuel jet if the fuel density is 760 kg/m3. For air c_p = 1.005 kJ/kgK and R = 287 J/kgK. Assume C_{df} = 1.0.

[IAS, 2005]

Solution

Given: $n = 2$, $V_s = 2$ litres or 2×10^{-3} m^3, $N = 4000$ rpm, $\eta_{vol} = 0.75$, A/F = 14, $d_2 = 30$ mm, $C_{da} = 0.9$, $C_{df} = 1.0$, $p_1 = 1.0$ bar, $T_1 = 293$ K, $\rho_f = 760$ kg/m^3, $c_p = 1.005$ kJ/kgK, $R = 287$ J/kgK

Actual volume sucked by the engine is, $V_1 =$

Air velocity at venturi throat,

or $p_2 = 0.964$ bar

$$\Delta p = p_1 - p_2 = 1 - 0.964 = 0.036 \text{ bar}$$

Mass of air flowing through the venturi/s,

$\therefore \dot{m}_f = 0.0408/14 = 2.916 \times 10^{-3}$ kg/

s

or $d_f = 1.26 \times 10^{-3}$ m or 1.26 mm

Example 10.9

An engine fitted with a single jet carburettor having a jet diameter of 1.25 mm has a fuel consumption of 6 kg/hr. The specific gravity of fuel is 0.7. The level of fuel in the float chamber is 5 mm below the top of the jet when the engine is not running. Ambient conditions are 1 bar and 17°C. The fuel jet diameter is 0.6 mm. The discharge coefficient of air is 0.85. Air-fuel ratio is 15. Determine the critical velocity of

flow at throat and the throat diameter. Express the pressure at throat in mm of water column. Neglect compressibility effect. Assume discharge coefficient of fuel flow is 0.60.

Solution

Given: $d_j = 1.25$ mm, $\dot{m}_j = 6$ kg/hr, $S_f = 0.7$, $z = 5$ mm, $p_1 = 1$ bar, $T_1 = 273 + 17 = 290$ K, $d_f = 0.6$ mm, $C_{da} = 0.85$, A:F = 15, $C_{df} = 0.60$.

$$pV = mRT$$
$$\rho_f = S_f \rho_w = 0.7 \times 10^3 = 700 \text{ kg/m}^3$$

Applying Bernoulli's equation for air flow at entrance and venturi throat, we have

Mass of air flow per second,

For the flow of fuel, we have

or

Mass of fuel flow per second,

Critical velocity of flow at throat,

$$\dot{m}_f = C_{df}\,\rho_f\,A_f\,c_f$$

or

or $c_f = 14.035$ m/s

Now

or $p_2 = 0.31$ bar

Again $\dot{m}_a = C_{da}\, \rho_a\, C_{a2}\, A_2$

or

or $d_2 = 9.6$ mm

Pressure at throat = p_2 / ρ_w ghw

or

10.13 ❒ FUEL INJECTION SYSTEMS IN SI ENGINES

Although the modern carburettor is cheap and reliable, it has several inherent disadvantages that make the supply of correct A:F mixture always difficult. The problem is further accentuated when a single carburettor has to supply the mixture to a multi-cylinder engine. In addition, the throat

restricts the flow of air to the engine and the maximum power. The maximum power can be increased by using a large throat as this affects the economy at low speed because of low air speed and worst control on the fuel spray and atomisation. Multiple carburettors assist the distribution problem but they also increase fuel consumption due to low air velocity at low engine speed and capital cost.

For these reasons, many attempts have been made to design a satisfactory petrol injection system, in which each cylinder is supplied with its correct quantity of petrol for each working cycle under all operating conditions. It is also claimed that the system devised to do this, allow

the engine to produce more power and gives less vapourisation troubles.

The advantages of fuel injection system over the conventional carburettor have been appreciated for many years. The quest for improved engine performance linked with fuel economy with legislation regarding to control of exhaust emission has enormously increased the use of fuel injection system. The incorporation of electronic control systems has also considerably helped the development of efficient and commercially viable systems.

There are two different injection systems for SI engines which are mechanically operated as follows:

1. Combustion chamber injection
2. Continuous port injection

The combustion chamber injection system is just similar to CI engines. However, this system is not adopted as today's emission and fuel economy requirements make it impractical.

10.13.1 Continuous Port Injection System (Lucas Mechanical Petrol Injection System)

Figure 10.34 shows a simplified line diagram of the aforementioned system. The petrol is sucked from the tank by a pump and pressurised petrol at 3 bar is supplied through a distributor to the fuel injector to a particular cylinder. The relief valve shown in figure maintains the pressure and allows excess petrol to return to the tank. The pump may be driven by engine or by engine or by a

separate electric motor. The latter is preferred because it starts pumping at its normal running speed and pressurises as soon as ignition is switched on.

In this system, petrol is continuously injected into the inner port as shown in Fig. 10.34 at a varied rate. The distribution may be made by having a separate metering pump for each cylinder, timed by the arrangement of a series of cams on one cam shaft or by having one pump operated by a single cam with a lob for each cylinder (similar to ignition system) that feeds to distribution unit which passes the petrol to each inlet port in turn. The unit may be driven by shaft, chain, or V-belt.

Nowadays, electronically controlled fuel

injection systems are commonly used as they function rapidly and respond automatically to the change in manifold air pressure, engine speed, crankshaft angle, and many other secondary factors. The electronic control unit assesses data (manifold pressure, engine speed, crank angle) received from various sensing devices and then adjusts the A:F supply for the best performance of the engine.

Figure 10.34 *Lucas mechanical petrol injection system*

This system has to contain a means of supplying additional fuel for cold starting, during warming, and enriching the mixture during acceleration. Ice formation is virtually impossible with this system and the danger of

vapourisation is minimised because petrol is under pressure right up to the injection point.

10.13.2 Electronic Fuel Injection System

The amount of fuel supplied to the engine is controlled by three factors namely, injector cross sectional area, fuel pressure, and duration of injection. Injector orifice size is fixed for a particular design. Fuel pressure supplied is controlled by electric pump and regulator. Therefore, the only variable factor in electronic fuel injection is to control the period of injection. This is done by translating the data supplied to the electronic control unit from the various sensors into electric pulses which are in turn relayed to the

solenoids which operate the injectors thus determining the moment and period of injection.

The main parts of the system are the injector and electronic control unit.

Injectors

The solenoid operated fuel injector is shown in Fig. 10.35(a). It consists of a valve body and a needle valve to which the solenoid plunger is rigidly attached. The fuel is supplied to the injector under pressure from the electric fuel pump passing through the filter. The needle valve is pressed against a seat in the valve body by a helical spring to keep it closed until the solenoid winding is energised. When a current pulse is received from the electronic control

unit, a magnetic field builds up in the solenoid coil which attracts a plunger and lifts the needle valve from its seat. This opens the path to pressurised fuel to emerge as a finely atomised spray.

Electronic Control Unit

This unit is the heart of electronic injection system which is responsible for metering the quantity of fuel supplied to each cylinder. The unit contains a number of printed circuit boards on which, a series of transistors, diodes, and other electronic components are mounted. This makes the vital data analysing circuits responding to various input signals. After processing the input data, the power output circuits in the control unit generates current pulses

which are transmitted to the solenoid injectors to operate the injector for the required period.

Figure 10.35 *Fuel injection valve: (a) Solenoid operated valve, (b) Electronically controlled valve*

All the electrical units of this system are connected to a cable terminating in a 25-pole plug to the mating socket. Hence, as soon as the ignition switch is actuated, the fuel pump is switched on and the fuel injection system becomes operational. The arrangement of this system is shown in Fig. 10.35(b).

Advantages of Petrol Injection System

1. The fuel injection system can precisely match fuel delivery to engine requirements under all load and speed conditions. This reduces fuel consumption with no loss of engine performance.
2. The manifold in an injection system carries only air, so there is no problem of the air and fuel separation and design of manifold becomes simple.
3. Due to absence of venturi which abstracts the air passage, the petrol injection results in better volumetric efficiency and increased power.

4. Fuel injection system is relatively free from icing and surge when tilted, cornering, and braking.
5. Starting and acceleration is much simpler than the carburettor system.
6. It provides an identical A:F ratio mixture to all the cylinders and maintains better balancing. In addition, the engine can work more economically closer to its lean limit, whereas in a carburettor engine, one or two ratios richer than best economy mixture are to be used to compensate for driveability. This gives lower specific fuel consumption.
7. By maintaining a precise A:F ratio according to engine requirements, exhaust emissions are lowered. The improved air fuel flow in injection system also helps to reduce emission levels.

Disadvantages of Petrol Injection System

1. The major disadvantage is its high capital cost due to precise and complicated components of electronic circuit.
2. The maintenance of this system is difficult and costly as this system contains 4 times the components of mechanical system. A Junker 12 cylinder engine has 1576 parts against 433 parts used with 12 cylinder Mercedes carburettor system.
3. The weight of the mechanical injection system is considerably higher than that of a carburettor.
4. The mechanical injection system has many wearing parts such as camshaft, rotor, and so on.
5. The injection system generates more noise.

10.13.3 Rotary Gate Meter Fuel Injection System

We have seen that a conventional fuel injection system requires individual injector to inject fuel into each cylinder. The additional components increase the

overall cost compared to the carburettor system.

This new system, shown in Fig. 10.36, reduces cost because fuel is injected at a central location and then distributed to each cylinder. There is a rotary gate valve at the heart of the system, controlled by engine intake air to regulate fuel injection. As the volume of the air taken into the engine cylinder increases, the gate valve tips to increase the amount of fuel injected into the engine.

Incidentally, Robert Bosch Corporation, Germany, reported that this system helped in optimising fuel and distribution as the fuel is injected into the air stream at the point of highest

velocity.

Figure 10.36 *Rotary gate make fuel injection system*

10.14 ❒ FUEL INJECTION IN CI ENGINES

Unlike SI engines, the fuel in CI engines is supplied at a very high pressure, partly during the compression stroke and partly during the power stroke. The air is taken in during the suction stroke and compressed to a high pressure (30–70 bar) and high temperature (500°C–700°C) according to the compression ratio used (12–20). The high temperature of air at the end of the stroke is sufficient to ignite the fuel.

As the fuel is injected in a high pressure air, the pressure of fuel injected lies between 100–200 bar. During the process of injection, the fuel is broken

into very fine droplets. The droplets vapourise, taking the heat from the hot air and form a combustible mixture and start burning. As the burning starts, the vapourisation of fuel is accelerated as more heat is available. As the combustion advances, the amount of oxygen available for burning reduces and therefore, the heat release rate is reduced.

The period between the injection and ignition of fuel is known as delay period (ignition delay). This lies between 0.001 and 0.002 seconds, according to the speed of the engine. This counts for nearly 25° crank rotation. The whole performance of the engine is totally dependent on the delay period. The less

is the delay period, better is the performance of the engine.

The main functions of the injection system are as follows:

1. To supply the correct quantity of fuel to be injected as per the load of the engine and increase in speed for automobile engines.
2. To supply the fuel within a precisely defined period of the cycles.
3. To control the rate of fuel injection such that it should result in the desired heat release pattern.
4. To atomise the fuel into very fine particles.
5. To distribute the fuel uniformly in the combustion chamber of the engine and results in rapid mixing of fuel and air.
6. Injection starts and stops sharply. There should not be any after injection.

10.14.1 Types of Injection Systems

Injection system may be divided into two general types, as follows:

1. Air injection system
2. Airless or solid or mechanical injection

Air Injection System

It was first developed by Rudolf Diesel.

The arrangement of the system is shown in Fig. 10.37. In this system, air and fuel are injected into the cylinder during the supply of fuel. The required pressure of the air for injecting the fuel is about 70 bar or higher.

A fuel pump is driven by the engine itself. A cam shaft operates the fuel pump through a cam and the power required to rotate the cam shaft is taken from the main shaft of the engine with the help of gears and discharges a definite quantity of fuel into the injection valve as shown in Fig. 10.37. The injection valve is mechanically opened and high pressure air drives the fuel charge and some air into the combustion chamber. The amount of

fuel delivered is under the control of oil pump suction valve, which is operated by a governor.

The air pressure is raised to about 70 bar by a three-stage compressor (as shown in Fig. 10.37) providing intercooling. The compressor is also operated by the engine. The high-pressure air projects the fuel into the combustion chamber and atomises it. Nowadays, such systems are rarely used in diesel engines.

Figure 10.37 *Airless injection system*

The advantages and disadvantages of this system are as follows:

Advantages

1. It provides better atomisation and distribution of fuel.
2. As the combustion is more complete, the brake mean effective power (BMEP) is higher than with other types of injection systems.
3. It allows to use the inferior fuels.

Disadvantages

1. It requires a complicated mechanism to run the compressor.
2. The weight of the engine increases.
3. A part of the power is used to drive the compressor and the BHP of the engine is reduced.

Airless or Solid Injection

In this system, the fuel is supplied at a very high pressure (150 bar) from the fuel pump to the fuel injector from where it is injected to the combustion chamber with the help of an injector. The main parts of this system are the fuel pump and the fuel injector. The fuel pump is operated by a cam which is mounted on a cam shaft. The power required to operate the cam is taken from the engine crank shaft. Depending

on the location of the fuel pumps and injectors, and the method used to meter the fuel, solid injection may further be classified as follows:

1. **Common rail system:** In this system, there is a single high pressure pump, which supplies high pressure fuel to a common header (rail). The accumulator is connected to different cylinders by a separate fuel line through the fuel nozzle as shown in Fig. 10.38. The pressure in the accumulator is maintained constant with the help of a pressure relief valve. The fuel is supplied to each cylinder by operating the respective fuel valve with the help of a cam mechanism driven by the engine crank shaft.

Figure 10.38 *Common rail system*

The quantity of fuel injected and the timing of injection are controlled by fuel valve and not by the injection pump. The arrangement of the fuel valve (metering meter) and the fuel nozzle is shown in Fig. 10.38. This system uses spring loaded injection valves which open and close by mechanical means.

The pressure used in this system ranges between 110–300 bar, according to the compression ratio used for the engine design.

The advantages and disadvantages of the system are listed below.

Advantages

1. It fulfils the requirements of either the constant load with variable speed or constant speed with variable load.
2. Only one pump is sufficient for a multi-cylinder engine.
3. Variation of pump supply pressure will affect all the cylinders uniformly.
4. The arrangement of the system is very simple and maintenance cost is less.

Disadvantages

1. Very accurate design and workmanship are required.
2. There is tendency to develop leaks in the injection valve.

2. **Distributor system:** This system, like the common rail system, employs a single high pressure pump as shown in Fig. 10.39. The high pressure pump in this system is used for metering and compressing the fuel and then the fuel is delivered to the common rotating distributor. The fuel is supplied to each cylinder by the distributor. In every cycle, the injection strokes of the pump are equal to the number of cylinders. The quantity of fuel supplied and the timing of fuel supply are done by single plunger (main pump). Therefore, equal amount of fuel is supplied to each cylinder and at the same point in the cycle. The function of the distributor is merely to select the cylinder to receive the fuel.

Figure 10.39 *Distributor system*

The distributor block selects a particular cylinder according to the cam coming in contact with the distributor as shown in Fig. 10.39. The appropriate valve opens just before the beginning of injection and oil is supplied to the required

cylinder.

3. **Individual pump and nozzle system:** This differs from constant pressure injection both in design and operation of the pump and fuel injector. Each injector has a separate pump and the injector contains a spring-loaded hydraulically operated automatic plunger valve. No separate mechanism is required to operate it.

In this system, separate pumps, each (depending on the number of cylinder) driven individually or a single pump having four plungers in a common block may be used. In this case, the single pump is driven by the crank shaft through a single cam shaft having individual cam for each cylinder.

The arrangement of the individual pump system is shown in Fig. 10.40 with all pumps in one block, four plungers in one barrel and a common cam shaft.

The design of this type of pump must be accurate as the volume of fuel injected per cycle is 1/20,000 of the engine displacement at full load and 1/100,000 of the engine displacement during idling. The time allowed for injecting such a small quantity of fuel is very limited (about 1/450 second at 1500 rpm of the engine providing injection through a 20° angle). The pressure requirements vary from 100–300 bar.

A comparison of various fuel injection systems is

given in Table 10.6.

Figure 10.40 *Individual pump system*

Table 10.6 *Comparison of fuel injection systems*

10.14.2 Design of Fuel Nozzle

The fuel injection into the cylinder through the fuel nozzle is shown in Fig. 10.41.

Figure 10.41 *Fuel injection through an injection nozzle*

Let,

$p_1 = p_{inj}$ = injection pressure, kN/m2

$p_2 = p_{cyl}$ = combustion chamber pressure, kN/m^2

ρ_f = density of fuel, kg/cm^3

τ = period of injection, s

Q = fuel sprayed, cm3/cycle/cylinder

d_f = diameter of injector

Let u_f = specific volume of fuel (assumed incompressible)

c_1 = velocity at section $1 - 1'$

c_2 = velocity at section $2 - 2'$

Thus,

Neglecting c_1, being very small compared with c_2,

where C_{df} = co-efficient of discharge for the fuel injector.

Duration of injection in second,

Mass of fuel supplied per second,

Volume of fuel injected per second

where d = orifice diameter

n = number of orifices

θ = crank angle during which the fuel is supplied

N_i = number of injections per minute

N = rpm

Fuel consumed per hour, m_f = power in

kW × SFC in kg/kWh.

Fuel consumed per cylinder per cycle (for 4-stroke engine),

Volume of fuel injected per cylinder per cycle,

Injection period

Example 10.10

Calculate the diameter of the injector orifice of a six-cylinder, 4-stroke CI engine using the following data:

Brake power = 250 kW; Engine speed = 1500 rpm; BSFC = 0.3 kg/

kWh; Cylinder pressure = 35 bar; Injection pressure = 200 bar; Specific gravity of fuel = 0.88; Co-efficient of discharge of the fuel orifice = 0.92; Duration of injection = 36° of crank angle.

[IES, 2007]

Solution

Given data: BP = 250 kW, N = 1500 rpm, BSFC = 0.3 kg/kWh, p_1 = 200 bar, p_2 = 35 bar, Specific gravity of fuel = 0.88, C_{df} = 0.92, Duration of injection = 36° of crank angle, Number of cylinders = 6, Number of strokes = 4

Fuel consumed per hour = BP × BSFC = 250 × 0.3 = 75 kg/h

Fuel consumed per cylinder per cycle

Volume of fuel injected per cylinder per cycle

Density of fuel, $\rho_f = 1000 \times 0.88 = 880$ kg/m^3

Nozzle hole area,

Diameter of cylinder orifice, $d_f = 0.75$ mm.

Example 10.11

A six-cylinder, four-stroke oil

engine develops 200 kW at 1200 rpm and consumes 0.3 kg/kWh. Determine the diameter of a single orifice injector if the injection pressure is 200 bar and combustion chamber pressure is 40 bar. The injection is carried for 30° rotation of a crank. Take $\rho_f = 900$ kg/m^3, and $C_{df} = 0.7$. Each nozzle on a cylinder is provided with a single orifice.

Solution

or $A_f = 0.0476 \times 10^{-4}$ m^2 or 4.76 mm^2

or $d_f = 1.005$ mm

Example 10.12

A 16-cylinder diesel engine has a power output of 800 kW at 900 revolutions per minute. The engine works on the four stroke cycle and has a fuel consumption of 0.238 kg/kWh. The pressure in the cylinder at the beginning of injection is 32.4 bar and the maximum cylinder pressure is 55 bar. The injector is set at 214 bar and maximum pressure at the injector is around 600 bar. The specific gravity of the fuel is 0.86. Calculate the orifice area required per injector, if the injection takes place over 10 degree crank angle.

[IES, 2001]

Solution

Given: $i = 16$, $P_t = 800$ kW, $N = 900$ rpm, $n = 2$, $\dot{m}_f = 0.238$ kg/kWh, $p_c = 32.4$ bar, $p_{max} = 55$ bar, $p_i = 214$ bar, $s = 0.86$, $\theta = 10°$, $(p_{i\max}) = 600$ bar

Power per cylinder, $P =$

Fuel consumption per cylinder, $m_f = \dot{m}_f \times P = 0.238 \times 50 = 1.19$ kg/h or 3.3056×10^{-3} kg/s

Fuel to be injected per cycle,

Fuel injection time,

Fuel mass rate per second,

Pressure difference in the beginning,
$\Delta p_i = p_i - p_c = 214 - 32.4 = 181.6$ bar

Pressure difference at the end,

$$\Delta p_e = (p_i)_{max} - p_{max} = 600 - 55 = 545 \text{ bar}$$

Average pressure difference,

Now $m_f = C_d A_i[2\rho_f \, \Delta p_a]^{1/2}$

or $0.238 = 0.6 \times A_i \, [2 \times 0.86 \times 10^3 \times 363.3 \times 10^5]^{1/2}$

or $A_i = 1.587 \times 10^{-6}$ m^2 or 1.587 mm^2

Example 10.13

An 8-cylinder, 4-stroke diesel engine has a power output of 368 kW at 800 rpm. The fuel consumption is 0.238 kg/kW-hr.

The pressure in the cylinder at the beginning of injection is 35 bar and the maximum cylinder pressure is 60 bar. The injector is adjusted to operate at 210 bar and the maximum pressure in the injector is set at 600 bar. Calculate the orifice area required per injector if the injection takes place over 12° crank angle. Assume the coefficient of discharge for the injector = 0.6, specific gravity of fuel = 0.85 and the atmospheric pressure = 1.013 bar. Take the effective pressure difference to be the average pressure difference over the injection period.

[IAS, 2004]

Solution

Given: $i = 8$, $n = 2$, BP = 368 kW, $N = 800$ rpm, $\dot{m}_f = 0.238$ kg/kWh, $p_1 = 35$ bar, $p_{max} = 60$ bar, $p_i = 210$ bar, $(p_i)_{\max} = 600$ bar, $\theta = 12°$, $Cd_f = 0.6$, $S_f = 0.85$, $p_{\text{atm}} = 1.013$ bar, $\Delta p_e = (\Delta p_{avg})_i$

Fuel consumed by the engine per cylinder =

Fuel consumed per cycle =

Duration of injection =

Rate of fuel consumption

Density of fuel $\rho_f = 0.85 \times 10^3 = 850$ kg/m^3

or $A_f = 2.49 \times 10^{-6}$ m^2 or 2.49 mm^2

or $d_f = 1.78$ mm

10.15 ❐ FUEL IGNITION

The ignition of fuel is concerned only with starting the combustion and not with the behaviour of the combustion flame. For starting the burning of fuel, it is necessary to raise the temperature of the air-fuel mixture to its ignition temperature. The energy required for this purpose is supplied through an electric spark. Within the range of mixtures normally used (12:1 to 15:1) in SI engines, a spark energy of 10 kJ is sufficient to start the combustion process.

10.15.1 Requirement of Ignition System

The important requirements of a spark

ignition system are as follows:

1. The voltage across the spark plug electrodes should be sufficiently large to produce an arc required to initiate the combustion. The voltage necessary to overcome the resistance of the spark gap and to release enough energy to initiate the self-propagating flame front in the combustible mixture is about 10,000–20,000 volts.
2. The intensity of the spark should lie in a specified limit because extremely high intensity may burn the electrodes and extremely low intensity may not ignite the mixture properly.
3. The volume of the mixture (clearance volume) at the end of compression should not too large, otherwise the spark produced may not be sufficient to ignite the whole charge. There is definite relation between the size of the spark and clearance volume.
4. There should be no missing cycle due to failure of spark.
5. In a multi-cylinder engine, there must be arrangement (distributor) to carry this voltage to the right cylinder at the right time.

10.15.2 Ignition Systems

The basic ignition systems in use are as follows:

1. Battery ignition system—conventional, transistor-assisted
2. Magneto ignition system—low temperature, high temperature
3. Electronic ignition system

Battery and magneto ignition systems differ only in the source of electrical

energy. A battery ignition system uses a battery, whereas a magneto ignition system uses a magneto to supply low voltage, all other system components being similar.

Battery Ignition System

The function of battery ignition system is to produce high voltage spark and to deliver it to the spark plugs at regular intervals and at the correct time with respect to the crank position. The required components of the system are as follows:

1. A battery of 6–12 volts
2. Ignition coil
3. Contact breaker
4. Condenser
5. Distributor
6. Spark plug

The arrangement of all the components

of battery ignition system for 4-cylinder engine is shown in Fig. 10.41.

The source of current is the storage battery and it is connected to the primary induction coil through the starting switch as shown in Fig. 10.42. The other end of the primary coil is connected to the breaker, which is connected to the ground, when the breaker contact points are closed. (In Fig. 10.42, the breaker contact points are shown in the open position). As one terminal of the battery is grounded, the circuit is closed by passing the current from the battery through the starting switch, primary coil, contact breaker, ground and back to the battery when contact points are closed.

The induction coil consists of primary winding, usually 100–200 turns and a secondary winding, usually 10,000 turns. Both windings are mounted on soft iron core.

The contact breaker consists of contact points, a camshaft on which a cam is mounted which is used to break and make the contacts between the contact points.

The distributor consists of a distributor arm, as shown in Fig. 10.42. The arm is mounted on a cam shaft and is rotated at half the speed of crankshaft. The function of the arm is to make the contact with each spark plug as shown in Fig. 10.42.

Figure 10.42 *Battery ignition system for multi-cylinder engines*

The distributor unit generally includes contact breaker to make the unit more compact, as both are driven by the same cam shaft.

A condenser is included in the circuit as shown in Fig. 10.42.

Principle of Induction

An EMF is produced in the coil due to the relative movement of magnetic lines and coil because the magnetic field lines are cut by the coil. The EMF produced depends on the relative movement between the two; higher the movement, greater the EMF.

The principle of induction from one coil into another is shown in Figs 10.43(a) to

(c). When current is allowed to flow through the primary coil, a magnetic field is set up and this field passes through the secondary coil and induces EMF, sending a current through a closed circuit. The current in the primary coil quickly attains a steady value and a magnetic field is stabilised. The EMF is not induced in the secondary coil as there is no relative movement between field (established by primary) and secondary coil.

If the primary is switched off, the established magnetic field collapses and the EMF is induced again in the secondary coil in the opposite direction. The greater EMF is induced when the circuit is broken because the collapse is

more rapid.

The EMF induced in the secondary coil can be further increased by collapsing the established field more rapidly. The magnetic effect in the secondary coil (EMF-produced) is intensified by winding the primary and secondary coils on a common soft iron bar or a ring. The lines of the magnetic field are concentrated around the bar or ring and the magnetic effects are intensified. A higher EMF at the secondary terminal can be obtained by suitable proportioning of primary and secondary coil around a common iron coil (1:100).

Figure 10.43 *Principle of induction*

Principle of Ignition

High voltage can be introduced at the

terminals of the secondary coil by collapsing the field established by primary and proper proportioning of the turns of primary and secondary as mentioned earlier. If this is connected to the two points providing an air gap between them as shown in Fig. 10.44, a spark is produced. Ignition can take place in the compressed charge of petrol engine if the spark is produced in the charge at the end of compression.

Working of Battery Ignition System

When the primary circuit is closed by the contact breaker (shown in open position in Fig. 10.42) a current begins to flow through the primary coil and magnetise core of the coil. The EMF is induced in the secondary as the current

in the primary increases. The EMF induced in the secondary coil is proportional to the rate at which the magnetic flux increases. The EMF produced in the secondary coil due to the growth of current in the primary coil is not sufficient to produce a spark at the spark plug because the primary circuit has to establish the magnetic flux.

When the primary circuit is opened by the contact breaker, the magnetic field collapses. Electromotive force is induced in the secondary which is directly proportional to the rate at which the magnetic field of the core collapses which in turn depends on the rate of decrease of the primary current. A condenser is connected across the

contact breaker in the primary circuit as shown in Fig. 10.42. This helps to collapse the field very rapidly by absorbing part of the energy of the magnetic field which is thrown back into the primary winding and produces a very high voltage in the secondary. This EMF in the secondary is sufficient to ignite the charge by producing the spark.

Figure 10.44 *Principle of ignition*

One end of the secondary coil is connected to the ground and the other end is connected to the central terminal of the distributor. The distributor connects the secondary coil in turn to the different spark plugs of the engine in their firing order. The spark plug of a

particular cylinder is placed in circuit of the secondary coil with the help of the distributor when the time comes for the charge in that cylinder to be ignited and at the same time the primary circuit is opened by contact breaker. A spark is produced between the points of the spark plug.

The distributor and contact breaker are generally mounted on the same cam-shaft which rotates at half speed of the crankshaft. The function of the distributor is to connect the secondary coil to each cylinder of a multi-cylinder engine at the time of ignition. The contact breaker also works simultaneously with the distributor and its function is to disconnect the primary

circuit exactly at the same time when the spark in the particular cylinder is required. The distributor arm connects four spark plugs in one rotation of the cam shaft and therefore, four contact points are required in four cylinder engines. The contact breaker has to break the contacts four times in four cylinder engines and requires four cams as shown in Fig. 10.42. If there are *n* cylinders, the contact points and cams required are also *n* in number.

In a single cylinder engine, the distributor is not required as in scooter engine, and single cam is sufficient for giving the spark. An ignition system used in single cylinder petrol engine is shown in Fig. 10.45. Instead of the

battery, the magneto is used in this system.

The cam is mounted on the crankshaft only as breaking of circuit during each rotation is required in two stroke engine and there is no necessity of cam-shaft.

Figure 10.45 *Battery ignition system for single cylinder engine*

Number of Sparks

The number of sparks produced must be equal to the number of working strokes in a single cylinder engine. If there are N_c, cylinders, the number of sparks produced for that engine are as follows:

$$N_s, \text{(Number of spark)} = n \times N_c$$

where n = number of working strokes and N_c = number of cylinders

Further, n = for 4-stroke cycle engine

= N for 2-stroke cycle engine where n is the rpm of the engine

Thus, for 4-stroke engine, N_s = and for 2-stroke engine, $N_s = NN_c$.

Advantages of Battery Ignition System

1. Its initial cost is low compared with magneto. This is the main reason for the adoption of coil ignition on cars and commercial vehicles.
2. It provides better spark at low speeds of the engine during starting and idling. This is because the maximum current is available throughout the engine speed range, including starting.
3. The maintenance cost is negligible, except for the battery.
4. The spark efficiency (intensity) remains unaffected by advance and retard positions of the timing control mechanism.
5. The simplicity of the distributor drive is another factor in favour of coil ignition.

Disadvantages of Battery Ignition System

1. The engine cannot be started if the battery runs down.
2. The weight of the battery ignition system is greater than the magneto which is a major consideration in adopting the system in aero-engines.
3. The wiring involved in the coil ignition is more complicated than the one used in a magneto ignition. This results in more likelihood of defects occurring in the system.
4. The sparking voltage drops with increasing speed of the engine.

Components of a Battery Ignition System

1. **Battery and cut-out:** A battery is an electro-chemical device which supplies current because of chemical reactions that occur in it. The common type of battery used in automobiles is a lead acid storage battery. A 12 V battery has six cells each generating two volts. Each cell consists of group of positive and negative plates. The positive plate has a grid of lead and antimony alloy filled with lead peroxide. The negative plate has a grid filled with spongy lead. The positive and negative plates are immersed in an electrolyte of dilute sulphuric acid. The plates are separated from each other by PVC or rubber separators. The positive and negative plate grids are connected to lead antimony strips. These strips separately connect the positive and negative sparks in series and the positive and negative terminals. The plates are placed in a battery container made of hard rubber which is acid-proof. The plate tops are supported by insulating and acid-resisting cell covers. Holes are provided in each cell to fill and unfill electrolytes or distilled water. The filter holes are covered with plugs containing small holds that allow gases to escape.

   The battery is charged continuously by a dynamo directly mounted on the crankshaft. To avoid over charging or discharging of the battery, an electric switch is introduced between the battery and dynamo, which is known as a cut-out.

2. **Ignition coil:** The purpose of the ignition coil is to step up 6 V or 12 V battery to 5000 V, which is sufficient to generate the spark. This coil consists of two insulated conducting coils having primary and secondary windings. The primary winding is connected to the battery through an ignition switch and contact breaker and the secondary winding is connected to the spark plugs through the distributor.
3. **Contact breaker:** As the number of sparks produced increases with an increase in engine speed, the time available for building up the magnetic field of the spark coil becomes shorter. As a result, the maximum value attained by the primary coil is lessened with increasing speed and the magnetic field produced becomes weaker until there is not enough voltage induced in the secondary winding to produce the spark. Contact breaker is a device to increase the time

during which the primary circuit remains closed.

4. **Condenser:** For obtaining the highest voltage in the secondary circuit, a quick collapse of the magnetic field is essential. In addition, it is also necessary to prevent the arcing and consequently burning of contact points.

This is achieved by providing a condenser. The condenser is a device which will absorb and hold an electric charge when an electromotive force is applied to its terminal. In its simplest form, it consists of two sheets of conducting material separated by a layer of insulating material.

The condenser's capacity is given by $C = A \times x \times p$, where A is the area of conducting material, x is the distance between the sheets of conducting material and p is the specific inductive capacity of the insulating material. The parts of the condenser are shown in Fig. 10.46.

In order to get large capacity with limited space, instead of two sheets of conducting material, two sets of such sheets are used and are separated by an insulating material. The conducting material used is 'tinfoil', whereas the insulating material is mica in condenser used with magneto and wax-impregnated paper for battery system. All conducting sheets of a set are electrically connected. The arrangement for the condenser is shown in Fig. 10.47.

The operation of the condenser is described as

follows. The condenser is connected across the contact breaker. When the contacts points open, instead of passing across the points in the form of an arc, the current flows into the condenser, is stored by it, and becomes charged. The charge in the condenser discharges back immediately in the primary circuit in the direction reverse to the flow of a battery current, thus assisting in a quicker collapse of magnetic field when the contact points open.

Figure 10.46 *Condenser*

Figure 10.47 *Condenser arrangement*

5. **Distributor:** The main function of the distributor is to distribute the high voltage surge to different plugs of multi-cylinder engine at the right time. The high tension current first goes to the control electrode of the distributor and then to the rotor. As the rotor rotates, it passes this high tension current to metal electrodes which are connected to the spark plugs through high tension wiring according to the firing order of the engine.

These are two types of distributors—brush type and gap type. The blow-up of distributor parts is shown in Fig. 10.48. In the brush type, a carbon brush carried by a rotor arm slides over a metallic segments embedded in the distributor cap of moulded insulating material, thus establishing electrical connection between the secondary winding of the coil and the spark plug.

In the gap type, the electrode of the rotor arm

passes close to, but does not actually contact the segments in the distributor cap. There is no appreciable wear and misfiring due to fouled spark plug.

When the surface of the spark plug insulator inside the combustion chamber becomes fouled with conducting carbon and oxides, there is considerable leakage of current through this conducting layer which prevents the voltage in the secondary from building up.

In actual practice, the distributor, contact breaker, rotating shaft with breaker cam, condenser, and the ignition advance mechanism are housed together.

6. **Spark plug:** The function of the spark plug is to generate the spark in the combustion using a high voltage communicated by the secondary. The spark plug provides two electrodes with a proper gap across which high potential is discharged and spark is generated.

A sectional view of a conventional spark plug is shown is Fig. 10.49. It consists of a steel shell, an insulator, and two electrodes. The high voltage supply from secondary is given to the central electrode which is insulated with porcelain. The other electrode is welded to the steel shell of the plug and thereby automatically grounded when the plug is fitted in the cylinder head of the engine. The electrodes are made of

high nickel alloy to withstand severe corrosion and erosion to which they are subjected.

Figure 10.48 *Distributor, blow up*

Figure 10.49 *Schematic of a typical spark plug*

The tips of central electrode and insulation are exposed to the burned gases. This results in high thermal stresses and the insulator may crack. As the tips are subject to high temperature (2000°C – 25.000°C), heat must flow from the insulator and tip to the surrounding shell in order to cool the electrodes and prevent pre-ignition.

The spark plugs are classified as hot plug and cold plug, depending on the temperature at the tip of the electrodes. The operating temperature of the tip depends on the amount of heat transferred, which, in turn, depends on the path followed by the heat to flow. A cold plug has a short heat flow path, whereas a hot plug follows a long flow path for the heat to flow as shown in Fig. 10.50.

The hot plug is used to avoid cold fouling where combustion chamber temperatures are relatively low such as during low power operation and continuous idling.

The temperature may be less at idling speed and

then tips will be fouled by unburned carbon deposits or excess lubricating oil. The carbon deposits burn at 350°C, whereas lubricating deposits burn at 550°C. If the spark plug runs hot at idling speed to prevent carbon deposits, it may run too hot at a high speed. This may cause undesirable pre-ignition. If the plug runs above 800°C, pre-ignition generally occurs.

The insulator tip length is the most important parameter which controls the operating temperature. Therefore, the tip temperature is generally controlled by varying the insulator tip position and the electrode material.

It is necessary in practice to compromise to obtain a proper spark plug which would operate satisfactorily throughout the engine operating range. An improper spark plug has remained a major source of engine trouble as misfiring and pre-ignition.

The factors affecting the operation of spark plug are (i) compression ratio, (ii) electrode temperature, (iii) speed of the engine, (iv) mixture strength, and (v) run of the engine.

Magneto Ignition System

Some of the drawbacks of the battery ignition system are as follows:

1. There are chances of discharging the battery.
2. There are chances of misfiring at higher speed of the engine.
3. It requires complicated wiring.
4. There are chances of failure.
5. There are many mechanical complications in the operation of the system.

These difficulties experienced with battery ignition system can be avoided by using magneto ignition system.

Figure 10.50 *Heat transfer path of hot and cold spark plug*

Magneto Ignition System—Working

The arrangement of the magneto ignition system is shown in Fig. 10.51. The only difference between the battery and the magneto system is that the battery is replaced by the rotating magnet. As the magneto rotates, the voltage and current are generated in the

primary coil and the circuit is completed by passing the current through the contact breaking point and the ground. As the current passes through the primary coil through the contact breaker, the circuit is completed through the ground. As the camshaft rotates, the cam 1 opens the contact breaker and interrupts the flow of current in the primary. This causes the decay in the magnetic field lines and cuts the lines of magnetic field in the secondary coil, and a high voltage is generated in the secondary coil. The process of generating the high voltage in the secondary coil is known as the induction phenomenon. The voltage generation in the secondary depends on the ration of number of turns in the secondary and

primary coils.

Further, consider the effect of a firing sequence on engine cooling. When the first cylinder is fired, its temperature increases. When the next cylinder fired is number 2, the portion of the engine between the cylinder number 1 and 2 gets overheated. If the third cylinder fired is cylinder number 3, this overheating is shifted to the portion between the cylinders 2 and 3. Thus, the task of the cooling system becomes very difficult because it is required to cool more at a place than at other places and this place too changes its position with time. If we fire the third cylinder after the first, the overheating problem can be mitigated.

Next, consider the flow of exhaust gases in the exhaust pipe. After firing the first cylinder, exhaust gases flow out to the exhaust pipe. If the next cylinder fired is the cylinder number 2, we find that before the gases exhausted from the first cylinder go out to the exhaust pipe, the gases exhausted from the second cylinder overtake them. This would require that the exhaust pipe be made bigger. Otherwise, the back pressure in it would increase and the danger of back flow will arise. If, instead of firing cylinder number 2, cylinder number 4 is fired, by the time the gases exhausted by cylinder 4 come into the exhaust pipe, the gases from cylinder 1 will have sufficient time to travel the distance between 1 cylinder and 4 and thus, the

development of a high back pressure would be avoided.

Figure 10.51 *Magneto ignition system*

It should be noted that to some extent, all the three requirements are conflicting and therefore, a trade-off is necessary. For four cylinder engines, the possible firing orders are 1-2-4-3 and 1-3-4-2. The latter is in common use. For a six-cylinder engine, the firing orders which can be used are 1-5-3-6-2-4, 1-5-4-6-2-3, 1-2-4-6-5-3, and 1-2-3-6-5-4. The first one is in common use.

Other Firing Orders

For three-cylinder engine 1-3-2

Eight-cylinder in-line engine

1-6-2-5-8-3-7-4

Eight-cylinder *V*-shape engine
1-5-4-8-6-3-7-2, 1-8-4-3-6-5-7-2, 1-6-2-5-8-3-7-4, 1-8-7-3-6-5-4-2, 1-5-4-2-6-3-7-8. Cylinder no. 1 is taken from front of the in-line engines, whereas in *V*-shape, front cylinder on right side-bank is considered cylinder no. 1 for fixing H.T. leads according to engine firing order.

Ignition Timing

In order to obtain maximum power from an engine, the compressed mixture must deliver its maximum pressure at a time when the piston is about to commence its downward stroke. Since there is a time lag between the occurrence of spark and the burning of the mixture, the

spark must take place before the piston reaches TDC on its compression stroke. Usually, it should occur at about 15° *b*TDC. If the spark occurs too early, combustion will take place before combustion stroke is completed and the pressure so developed will oppose the piston motion and thereby, reduce the engine power. If the spark occurs too late, the piston will have already completed a certain part of the expansion stroke before the pressure rise occurs and a corresponding amount of engine power will be lost.

The correct instant for the introduction of spark is mainly determined by the ignition lag. The ignition lag depends on many factors such as compression ratio,

mixture strength, throttle opening engine temperature, combustion chamber design, and speed. Some of the important factors affecting the ignition timings are as follows:

1. **Engine speed:** When an engine is rotating at 2000 rpm, even one-thousandth of a second represents 12° of crank rotation. Therefore, the spark must occur, say, 20° before the TDC for maximum power and economy. As the speed of the engine increases, the combustion time increases in terms of crank angle degrees and the spark must advance accordingly, that is, the angle of the advance must increase as the speed increases (see Fig. 10.52).
2. **Mixture strength:** In general, rich mixtures burn faster. Therefore, as the mixture is made richer, the optimum spark advance must be retarded, that is, the number of degrees of the crank angle before TDC is decreased and the spark occurs closer to the TDC.
3. **Part load operation:** Part-load operation of a spark ignition engine is affected by throttling. Due to throttling, a smaller amount of charge enters the cylinder and the dilution that occurs due to residual gases is greater. In addition, higher air-fuel ratio required for the part-load operation causes the combustion time to increase. Therefore, at part load, the spark advance must be increased.
4. **Type of fuel:** Ignition delay will depend on the type of fuel used in the engine. For maximum power and economy, a slow-burning fuel will need a higher spark advance than a fast-burning fuel.

It is obvious from the above that the point in the cycle where the spark occurs must be regulated to ensure maximum power and economy at different speeds and loads and must be

automatic. Most engines are fitted with a mechanism which is integral with the distributor and automatically regulate the optimum spark advance to account for change of speed and load.

Figure 10.52 *Combustion time in terms of engine speed and degrees of crankshaft rotation*

10.16 ❐ COMBUSTION IN IC ENGINES

Combustion is defined as a rapid chemical reaction between H_2 and C with oxygen in the air, and in the process, the reaction liberates energy in the form of heat.

It is absolutely essential to burn the fuel supplied completely for the economical working and safety of the engine. Therefore, the mixture supplied to the engine should possess the required A:F ratio; otherwise, combustion cannot be initiated or if initiated, it cannot be

sustained. In addition, there must be some means to initiate the combustion and the generated flame should be able to propagate through the mixture and burn the mixture completely.

It is a known fact that the fuel with any A:F ratio cannot be burned. There is an ignition limit for any fuel to start the combustion and sustain it till the complete fuel burns by the flame generated with spark plug. In addition, the temperature of the mixture to initiate the ignition is equally important. It is also known that the flame will propagate if the temperature of the burnt gases exceeds 1500 K for SI engine (gasoline) fuels. The ignition limits of the hydrocarbon fuel when temperature of

mixture reaches to 1500 K are shown in Fig. 10.53.

The upper and lower limits of A:F ratio for ignition depend on the temperature of a particular fuel. The limit becomes wider at higher temperatures because of higher reaction rate and higher thermal diffusivity coefficients of the mixture. Therefore, it is essential to see that the A:F ratio of the mixture supplied to the engine lies in the practical limit as shown in Fig. 10.53.

Figure 10.53 *Ignition limit of hydro-carbon fuels at 1500* K

10.16.1 Stages of Combustion in SI Engines

It is assumed that the heat is added instantaneously at constant volume in the ideal air-cycle of SI engines. To

achieve this, the burning of fuel in the SI engine should be instantaneous. In actual engines, combustion occurs over a finite period of time as the flame starting around the spark plug has to propagate through the entire mixture of air and fuel.

A high intensity spark is produced for a few degrees of crank angle before TDC at the end of the compression stroke. The function of the spark is to provide a source of heat to the combustible mixture. The temperature of the combustible mixture surrounding the spark is raised to a sufficient temperature level to start the combustion. The flame propagation is continued through the combustible

mixture, provided the release of heat from the initial sources is sufficient to heat the adjoining portion of the mixture to a temperature at which the heat from the reaction is sufficient to overcome the heat losses. The heat liberated by the burning portion ‘flame front’ prepares the adjacent portion of the unburned charge for the combustion reaction.

It has been observed experimentally that there is a certain time interval between the instant of spark and the instant when there is a noticeable rise in pressure due to combustion. This means that the combustion starts some time after the spark. This time lag is known as ‘ignition lag’. Ignition lag is a time interval in the process of chemical

reaction during which the molecules of the fuel get heated up to the self-ignition temperature, get ignited, and produce a self-propagating nucleus of flame.

The pressure variation in the SI engine combustion chamber during the crank rotation is shown in Fig. 10.54. This is an unfold *p*-*v* diagram.

The ignition is timed to take place at the point *a* but the burning commences only at the point '*b*'. The time interval between the points '*a*' and '*b*' is known as 'ignition lag'. This is generally expressed in terms of crank angle and is given by θ_1 as shown in Fig. 10.54. The time required for the crank to rotate through an angle θ_2 is known as combustion period during which the

propagation of flame takes place. Many times, the combustion is not complete at the point *c* (when the mixture is too rich) and the fuel may burn after the point *c* during the expansion stroke. This burning of fuel after the point of maximum pressure is known as ‘after burning’. This is not desirable.

The major disadvantages of the ignition lag is to reduce the power developed. If the ignition lag is long, the peak pressure occurs during the expansion stroke, and complete advantage of expansion is not achieved. The time lag may vary from 10°–50°, according to the type of the fuel used and several other factors affecting the time lag. To maintain the ‘time lag’ constant at

variable speed of the engine, it is necessary to change the angle of advance. The angle of advance must increase with increase in speed. This is commonly done in automobile engines by an automatic advancing mechanism.

Figure 10.54 *Crank angle v's pressure variation in SI engine*

All considerations should be taken into account in the design of the combustion chamber and selecting the fuel used to reduce the ignition lag.

The theoretical diagram of combustion is shown in Fig. 10.55(a).

However, the actual process differs from theory as instant combustion is not possible as shown by *bc* in Fig. 10.55(a). Some time (order of

millisecond) is needed to start the combustion after giving the spark because of the fact that the surrounding mixture is to be heated up to ignition temperature and then formed nucleus of flame which starts propagating through the surrounding mixture.

The pressure variation in the engine with respect to crank-angle is shown in Fig. 10.55(b). There are mainly three stages of combustion in SI engines as shown in Fig. 10.55(b).

1. **First phase:** This phase is considered between the point of ignition and point of combustion. As shown in Fig. 10.55(b), ignition is timed at point a and combustion starts at point b. The time interval between the points a and b is known as ignition lag. The ignition lag is generally expressed in terms of crank angle (θ_1). This period of ignition lag is very small and lies between 0.00015 and 0.002 seconds. An ignition lag of 0.002 seconds corresponds to 35° crank rotation when the engine is running at 3000 rpm and crank angle required (which is known as angle of advance) increases with an increase in engine speed.

Figure 10.55 *Pressure vs crank angle diagram: (a) Theoretical, (b) Actual*

2. **Second phase:** Once the flame is formed at point *b*, it should be self-sustained and be able to propagate through the mixture. This is possible when the rate of heat generation (Q_g) by burning the surrounding mixture of the flame nucleus is higher than the heat lost (Q_l) by the flame to the surrounding. As the difference between ($Q_g - Q_l$) is higher, the rate of flame propagation is higher and complete combustion will occur at the earliest, which is the most desirable requirement of combustion in SI engines. The propagation of flame also depends on the flame temperature and the temperature and density of the surrounding mixture as its propagation is directly proportional to these factors. A weak spark and a low compression ratio (as density of mixture is less) give low propagation of the flame.

After point *b*, the flame propagation is abnormally low in the beginning as heat lost is more than heat generated. Therefore, the pressure rise is also slow as the mass of mixture burned is small. Therefore, it is necessary to provide an angle of advance of 30° to 35° if the peak pressure is to be attained at 5°–10° after TDC.

After the point *c*, the pressure starts falling due to the fall in the rate of heat release when the flame reaches the wall in the last part of combustion and cannot compensate for its fall due to the gas expansion, and heat is transferred to the walls.

The time required for the flame to travel 95% of the chamber length with respect to speed of the engine is shown in Fig. 10.56, when $\theta_1 = 30°$ and A:F = 13.

It is obvious that the crank angle required for 95% travel increases with an increasing engine speed (the time available is decreased), therefore, if the combustion is to be completed at point *c*, the angle of advance (θ) must be increased with increasing engine speed. The flame speed increases with increasing engine speed because of the increase in turbulence of the mixture.

The time required for the crank to rotate through an angle θ_2 is known as *combustion period* during which the propagation of the flame takes place.

Stages I and II are not entirely distinct. The starting point of stage II is measurable as the rise in pressure can be seen on the p-θ diagram. This is the point where the line of combustion departs from the line of compression.

Figure 10.56 *Time required for the flame to travel 95% of the chamber length*

3. **Third phase:** Although the point c represents the end of the flame travel, it does not assure the complete combustion of fuel. In this case, the combustion still continues after attaining the peak pressure and is known as *afterburning*. This is continued throughout the expansion stroke. This generally happens when the rich mixture is supplied to the engine.

10.16.2 Ignition Lag (or Delay) in SI Engines

The ignition delay period is the phase during which some fuel has already been admitted but has not yet ignited. The fuel does not ignite immediately upon injection into the combustion chamber. There is a definite period of inactivity between the time when the first droplet of fuel hits the hot air in the combustion chamber and the time it starts through the actual burning phase. This period is known as the *ignition delay period*. In Fig. 10.57, the delay period is shown on the pressure crank angle (or time) diagram between points *a* and *b*. Point *a* represents the time of injection and point *b* represents the time at which the pressure curve first separates from the motoring curve.

The important factors affecting delay period are as follows:

1. **Compression ratio:** With increase in compression ratio, the minimum auto ignition temperature of a fuel decreases due to increased density of the compressed air. This results in closer contact between the molecules of fuel and oxygen, reducing the time of reaction. The increase in the compression temperature as well as the decrease in the minimum auto ignition temperature decreases the delay period. The peak pressure during the combustion process is only marginally affected by the compression ratio (because delay period is shorter with higher compression ratio and hence the pressure rise is lower).

   One of the practical disadvantages of using a very high compression ratio is that the mechanical efficiency tends to decrease due to increase in weight of the reciprocating parts. Therefore, in practice, engine designers always try to use a lower compression ratio which helps in easy cold starting and light load running at high speeds.

2. **Engine speed:** The delay period decreases with an increase in engine speed in a variable speed operation with a given fuel. With increase in engine speed, the loss of heat during compression decreases, resulting in the rise of both the temperature and pressure of the compressed air thus reducing the delay period.
3. **Power output:** With an increase in engine output, the air–fuel ratio decreases, operating temperatures increase, and hence, delay period decreases.

Figure 10.57 *Pressure-time diagram illustrating ignition delay*

4. **Quality of fuel:** A lower self-ignition temperature results in a

lower delay period. In addition, fuels with higher cetane number give lower delay period, and smoother engine operation. Other properties such as volatility, latent heat, viscosity, and surface tension also affect the delay period.

Table 10.7 shows the summary of the effects of variables on delay period.

Table 10.7 *Effect of variables on delay period*

10.16.3 Factors Affecting the Flame Propagation

After the ignition, the rate of flame propagation affects the combustion process in SI engines.

Better the propagation, higher is the combustion efficiency and higher the economy. The propagation depends on the flame velocity. Unfortunately, flame velocities for most fuels range between 10–30 m/s. Therefore, all steps should

be taken in the design and operation of the engine so that the flame velocity is as maximum as possible.

The study of flame propagation rate and the factors affecting the same is important for the following two reasons:

1. Its effect on the rate of pressure rise in the cylinder.
2. Its effect in connection with abnormal combustion (knocking).

The factors which affect the flame propagation are discussed below:

1. **A:F ratio:** The mixture strength influences the rate of combustion and the amount of heat generated. The maximum flame speed for all hydrocarbon fuels occurs at nearly 10% rich mixture. Flame speed is reduced for lean and rich mixture.

   Lean mixture releases less heat, resulting in lower flame temperature and lower flame speed. Very rich mixture results in incomplete combustion ($C \rightarrow CO$ instead of CO_2) and in production of less heat; the flame speed remains low.

   The effects of the A:F ratio on the *p-v* diagram

and p-θ diagram are shown in Fig. 10.58.

2. **Compression ratio (r):** A higher compression ratio increases the pressure and temperature of the mixture and also decreases the concentration of residual gases. All these factors which are in favour reduces the ignition lag and helps to speed up the second phase of combustion. The maximum pressure of the cycle as well as the mean effective pressure of the cycle will increase with an increase in r. Figure 10.59 shows the effect of the compression ratio on pressure (indirectly on the speed of combustion) with respect to crank angle for same A:F ratio and same angle of advance. Higher compression ration increases the surface to volume ratio and thereby increases the part of the mixture which afterburns in the third phase.

Figure 10.58 *Indicator diagrams for stoichiometric and weak mixtures*

Figure 10.59 *Effect of compression ratio on pressure rise*

3. **Load on engine:** With increase in load, the cycle pressures increase and the flame speed also increases.

In an SI engine, the power developed by an engine is controlled by throttling. At lower load and higher throttle, the initial and final pressure of the mixture after compression decrease and the mixture is also diluted by more residual gases. This reduces the flame propagation and prolongs the ignition lag. Therefore, the advance mechanism is also provided with a change in load on the engine. This difficulty can be partly overcome by providing rich mixture at part loads but this definitely increases the chances of after-burning. The after-burning is prolonged with richer mixture.

In fact, poor combustion at part loads and

necessity of providing richer mixture are the main disadvantages of SI engines which cause wastage of fuel and discharge of a large amount of CO with exhaust gases.

4. **Turbulence and engine speed:** Turbulence plays very important role in the combustion of fuel as the flame speed is directly proportional to the turbulence of the mixture. This is because turbulence increases the mixing and heat transfer coefficient or heat transfer rate between the burned and the unburned mixture. The turbulence of the mixture can be increased at the end of compression by a suitable design of the combustion chamber (geometry of cylinder head and piston crown).

   Insufficient turbulence provides low flame velocity and incomplete combustion and reduces the power output. However, excessive turbulence is also not desirable as it increases the combustion rapidly and leads to detonation. Excessive turbulence causes to cool the flame generated and flame propagation is reduced.

   Moderate turbulence is always desirable as it accelerates the chemical reaction, reduces ignition lag, increases flame propagation, and even allows the weak mixture to burn efficiently.

   The turbulence of the mixture increases with an increase in engine speed. Therefore, the effects of an increase in engine speed are similar to increase in turbulence.

5. **Other factors:** Among other factors, the factors which

increase flame speed include supercharging of the engine, spark timing, and residual gases left in the engine at the end of exhaust stroke.

The air humidity also affects the flame velocity but its exact effect is not known. Anyhow, its effect is not large compared with A:F ratio and turbulence.

Example 10.14

The spark plug is fixed at 18° before top dead centre (TDC) in an SI engine running at 1800 rpm. It takes 8° of rotation to start combustion and get into flame propagation mode. Flame termination occurs at 12°C after TDC. Flame front can be approximated as a sphere moving out from the spark plug which is offset 8 mm from the centre line of the cylinder whose bore diameter is 8.4 cm. Calculate the effective flame

front speed during flame propagation. The engine speed is increased to 34000 rpm and subsequently as a result of which the effective flame front speed increases at a rate such that it is directly proportional to 0.85 times of engine speed. Flame development after spark plug firing still takes 8° of engine rotation. Calculate how much engine rotation must be advanced such that the flame termination again occurs at 12° after TDC.

[IES, 2010]

Solution

Given that $\theta_1 = 8°$, $\theta_2 = 10° + 12° = 22°$, $N_1 = 1800$ rpm, $e = 8$ mm, $D =$

8.4 cm, $N_2 = 34000$ rpm,

The combustion process in p-v diagram is shown in Fig. 10.60.

Time taken,

Flame front speed,

When engine speed is increased to 34000 rpm, then

Figure 10.60 *Combustion process in p-v diagram*

10.16.4 Phenomena of Knocking/Detonation in SI Engines

In a SI engine, combustion is initiated between the spark plug electrodes and then spreads across the combustible mixture. A definite flame front which

separates the fresh mixture from the products of combustion travels from the spark plug to the other end of the combustion chamber. Heat release due to combustion increases the temperature and pressure of the burned part of the mixture above those of the unburned mixture. In order to effect pressure equalisation, the burned part of the mixture will expand and compress the unburned part adiabatically, thereby increasing the pressure and temperature of the unburned part further. If the temperature of the unburned mixture exceeds the self-ignition temperature of the fuel and remains at or above this temperature during the ignition lag, spontaneous ignition (or auto-ignition) occurs at various pin-point locations.

This phenomenon is called *detonation* or *knocking*.

The phenomenon of knocking may be explained by referring to Fig. 10.61(a) which shows the cross-section of the combustion chamber with flame advancing from the spark plug location A without knock, whereas Fig. 10.61(c) shows the combustion process with knock. In the normal combustion, the flame travels across the combustion chamber from *A* towards *D*. The advancing flame front compresses the end charge *BB'D* farthest from the spark plug, thus raising its temperature. The temperature is also increased due to heat transfer from the hot advancing flame front and the pre-flame oxidation in the

end charge. If the temperature of the end charge does not reach its self-ignition temperature, the charge would not auto-ignite and the flame will advance further and consume the charge $BB'D$. This is the normal combustion process which is illustrated by means of the pressure-time diagram in Fig. 10.61(b). If the end charge $BB'D$ reaches its auto-ignition temperature and remains for some period of time equal to the time of pre-flame reactions, the charge will auto-ignite, leading to detonation. In Fig. 10.61(c),it is assumed that when the flame has reached the position BB', the charge ahead of it has reached critical auto-ignition temperature. During the pre-flame reaction period, if the flame front could move from BB' to only CC',

the charge ahead of *CC'* would auto-ignite. Due to auto-ignition, another flame front starts travelling in the opposite direction to the main flame front. When the two flame fronts collide, a severe pressure pulse is generated. The gas in the chamber is subject to compression and rarefaction along the pressure pulse until pressure equilibrium is restored. This disturbance can force the walls of the combustion chamber to vibrate at the same frequency as the gas (≈ 5000 Hz). The pressure-time trace of such a situation is shown in Fig. 10.61(d).

Figure 10.61 *Normal and abnormal combustion: (a) Movement of flame front for normal combustion, (b) P-θ diagram for normal combustion, (c) Movement of flame front for abnormal combustion, (d) P-θ diagram for abnormal combustion*

10.16.5 Factors Influencing Detonation/Knocking

The factors influencing knocking are as follows:

1. **Density factors:** Any factor which reduces the density of the charge tends to reduce knocking by providing lower energy release.
    1. **Compression ratio:** Increase in compression ratio increases the pressure and temperature of the gases at the end of compression stroke which decreases the ignition lag of the end gas and increases the tendency of knocking.
    2. **Mass of induced charge:** An increase in the mass of induced charge into the cylinder of an engine by throttling or by increasing the amount of supercharging increases both temperature and density of the charge at the time of ignition. This increases the tendency for knocking.
    3. **Inlet temperature of mixture:** Increase in inlet temperature of mixture makes the compression temperature higher thereby, increasing the tendency of knocking.
    4. **Temperature of combustion chamber walls:** Hot spots (such as spark plug and exhaust valve) in combustion chamber promote knocking.
    5. **Retarding spark timing:** Having the spark closer to the TDC reduces knocking. However, this affects the brake torque and power output of the engine.
    6. **Power output of the engine:** A decrease in the output of the engine reduces the tendency to knock.
2. **Time factors:** The following factors reduce the possibility of knocking:
    1. **Turbulence:** Turbulence increases the engine speed and reduces the time available for the end charge to attain auto-ignition conditions thereby decreasing the tendency to knock.
    2. **Engine speed:** An increase in engine speed increases turbulence and decreases the tendency to knock.
    3. **Flame travel distance:** Shortening the time required for the flame front to traverse the combustion chamber reduces the tendency for knocking. Flame

travel distance is governed by combustion chamber size and spark plug position.

4. **Engine size:** A large engine (combustion chamber size) has a greater tendency for knocking as the flame requires a longer time to travel across the combustion chamber.
5. **Combustion chamber shape:** The combustion chamber should be such that it promotes turbulence to reduce knocking. Spherical chambers minimise knocking tendency.
6. **Location of spark plug:** A centrally located spark plug or multiple spark plugs minimise flame travel time and reduce knocking.

3. **Composition factors:**
    1. **Fuel:** F/A ratio affects the reaction time or ignition delay. When the mixture is nearly 10% richer than stoichiometric (F/A ratio = 0.08), ignition lag of the end gas is minimum, velocity of flame propagation is maximum, and knocking tendency is maximum. A rich mixture is effective in decreasing the knocking tendency due to a longer delay period and lower temperature of compression.
    2. **Octane rating of fuel:** Higher the octane number, lesser is the tendency for knocking. Paraffin series have the maximum and aromatic series the minimum tendency to knock. The knocking characteristics of a fuel can be decreased by adding small amounts of additives called dopes.
    3. **Humidity of air:** Increasing the humidity of the atmospheric air decreases the tendency to knock.
4. **Temperature factors:** Increasing the temperature of the unburned mixture by any factor in design or operation will increase the possibility of knocking in SI engine. The temperature of the unburned mixture is increased by the following factors:
    1. **Raising the compression ratio:** For a given engine setting and fuel, there will be a critical compression ratio above which knocking occurs. This compression ratio is called the highest useful compression ratio (HUCR).
    2. Supercharging
    3. Raising the inlet temperature
    4. Raising the coolant temperature

5. Increasing the load (opening the throttle)
6. Raising the temperature of the cylinder and combustion chamber walls
7. Advancing the spark timing

A summary of variable affecting knocking in SI engines is given in Table 10.8.

Table 10.8 *Summary of variables affecting knock in an SI engine*

10.16.6 Methods for Suppressing Knocking

1. **Proper location of spark plug:** A compact combustion chamber with proper central location of the spark plug reduces the path of the flame front travel from the spark plug to the remotest part of the combustion chamber, and thereby, reduces the tendency of knocking. The same result could be obtained by multi-fuel injection system.
2. **Proper selection of material for piston and cylinder head:** The end charge cooling is much better when the piston and cylinder are made of aluminium alloys as they have good thermal conductivity. A better cooling of the end charge reduces the tendency for knocking.
3. **Injecting water into the intake manifold:** Injection of water in the cylinder reduces the temperature of the end gas and increases the delay period, thereby decreasing the tendency to

knock.

4. **Retarding the spark timing:** By retarding the spark timing, the peak pressures are reached only during the power stroke and are of lower magnitude. This reduces knock.
5. **Extremely rich or lean mixture:** By using extremely rich or lean mixture, the flame temperature can be kept low, thus considerably eliminating the tendency of knock.
6. **Squish recesses in combustion chamber:** The provision of squish recess in combustion chamber cools the last position of the charge and reduces the tendency of knock.

10.16.7 Effects of Knocking/Detonation

1. **Mechanical damage:** Knocking creates a high pressure wave of large amplitude. This increases the rate of wear of almost all mechanical parts such as piston, cylinder head, and valves. The engine parts are also subjected to very high temperatures due to auto-ignition and the piston is damaged by overheating.
2. **Noise:** When the intensity of the knock is high, a loud pulsating noise is created due to high intensity pressure wave which vibrates back and forth across the cylinder. The high vibrating motion of the gases causes crankshaft vibrations and engine runs rough.
3. Heat transfer rate: The heat transfer rate increases as more heat is lost to the coolant.
4. **Power output:** The power output slightly decreases due to rapid burning of last part of the charge.
5. **Pre-ignition:** Pre-ignition is the ignition of the charge in the absence of spark as it comes in contact with hot surface. This adds negative work by the piston during compression which increases high peak pressure and temperature in the cylinder.

10.17 ❒ COMBUSTION CHAMBERS FOR SI ENGINES

The design of combustion chamber involves the shape of the combustion chamber, location of the spark plug, and disposition of inlet and exhaust valves.

10.17.1 Basic Requirements of a Good Combustion Chamber

The three main requirements of a SI engine combustion chamber are high power output with minimum octane requirement, high thermal efficiency, and smooth engine operation.

1. Higher power output requires the following:
    1. High compression ratio
    2. Small or no excess air
    3. No dead pockets
    4. An optimum degree of turbulence
    5. High volumetric efficiency
2. High thermal efficiency requires the following:
    1. High compression ratio
    2. Small heat loss during combustion
    3. Good scavenging of exhaust gases

3. Smooth engine operation requires the following:
    1. Moderate rate of pressure rise during combustion.
    2. Absence of detonation: compact combustion chamber, proper location of spark plug and exhaust valve, and satisfactory cooling of spark plug points.

10.17.2 Combustion Chamber Design Principles

1. To achieve high volumetric efficiency, the largest possible inlet valve should be accommodated with ample clearance around the valve heads.
2. To prevent detonation, the length of the flame travel from the spark plug to the farthest point in the combustion space should be as short as possible.
3. To reduce the possibility of detonation, the exhaust valve should be near the spark plug.
4. The exhaust valve should be kept small due to its hot surface, but to compensate for this, a high lift should be employed.
5. The correct amount of turbulence should be created in the combustion chamber by properly positioning the inlet valve. Turbulence should be preferably created by squish.
6. The shape of the combustion chamber should be such that the largest mass of the charge burns as soon as possible after ignition with progressive reduction in the mass of charge burnt towards the end of combustion.
7. To ensure high thermal efficiency and less air pollution, the surface–volume ratio of the chamber should be minimum in the beginning.
8. In the end gas region, surface–volume ratio should be large to achieved good cooling in the ‘detonation zone’.
9. The spark plug should be so positioned so that it will be scoured of any residual exhaust products by the incoming charge.
10. The exhaust valve head should be well-cooled as it is the hottest region of the combustion chamber.
11. The sparking plug points should be sufficiently cooled to avoid pre-ignition effects.
12. There should be good scavenging of exhaust gases.
13. Thickness of wells must be uniform for uniform expansion.
14. It is desirable to employ a plain flat-topped piston on manufacturing grounds.
15. To achieve maximum thermal efficiency, the highest possible

compression ratio should be used.

10.17.3 Combustion Chamber Designs

The various types of combustion chambers are shown in Fig. 10.62.

1. **T-head combustion chamber:** In the earliest engine, the T-head design shown in Fig. 10.62(a) was used. Its disadvantages are that it has two camshafts and it is prone to detonation.
2. **L-head or side valve combustion chamber:** This is shown in Fig 10.62(b). Its advantages are as follows:
    1. A side valve engine is easy to manufacture.
    2. It is easy to enclose and lubricate the valve mechanism.
    3. Its head can be removed without disturbing the valve-gear and pipe work for decarbonising.
    4. It provides a compact layout.
    5. Cooling of exhaust by inlet charge is more effective as both are situated by the side.

    Its shortcomings are as follows:

    1. There is lack of turbulence in the incoming charge.
    2. It is extremely prone to knocking as the flame has to travel long distance.
    3. It is extremely sensitive to ignition timing due to slow combustion.

Figure 10.62 *Examples of typical combustion chambers: (a) T-head type, (b) L-head type, (c) L-head side valve type, (d) I-head type, (e) F-head type*

3. **Ricardo turbulent L-head side valve design:** This arrangement is shown in Fig. 10.62(c). This gives higher flame speed and reduces the knocking tendency. The entire combustion space is concentrated over the inlet and exhaust valves. This provides sufficient turbulence during the compression stroke and increased flame velocity. The ratio of

area to volume of the space above the piston is large, which provides sufficient quench area essential to avoid knocking.

4. **Overhead valve or I-head combustion chamber:** In this arrangement, both the valves are located in the cylindrical head as shown in Fig. 10.62(d). This arrangement is superior to side valve or L-head arrangement when high compression ratios are used for the following reasons:
    1. Lower pumping losses and higher volumetric efficiency from better breathing of the engine from larger valves or valve lifts and more direct passage ways.
    2. Less distance for the flame to travel and therefore greater freedom from knock.
    3. Less force on the head bolts and therefore less possibility of leakage.
    4. Absence of exhaust valve from block results in more uniform cooling of cylinder and piston.
    5. Lower surface-volume ratio and, therefore, less heat loss and less air pollution.
    6. Easier to cast and hence lower casting cost.
5. **F-head combustion chambers:** A combustion chamber in which one valve is located in the head and other in the block is known as F-head combustion chamber, as shown in Fig. 10.62(e). This arrangement is a compromise between L-head and I-head combustion chambers. Figure 10.62(e) shows exhaust valve in head and inlet valve in block. This gives a rather poor performance. The modern F-head engines have inlet valve in the head and exhaust valve in the block. The valves are inclined with plug located in the flat roof. This provides the shortest flame travel and ensures minimum knocking tendency.

10.18 ❐ COMBUSTION IN CI ENGINES

In the CI engines, only air is compressed through a large compression ratio during the compression stroke, raising highly its temperature and pressure. At this

stage, one or more jets of fuel are injected in the liquid state, compressed to a high pressure by means of a fuel pump. Each minute droplet, as it enters the hot air, is quickly surrounded by an envelope of its own vapour and after an appreciable interval, is inflamed at the surface of the envelope. As soon as this vapour and the air in contact with it reach a certain temperature and the local air-fuel ratio is within combustion range, ignition takes place.

10.18.1 Stages of Combustion

The stages of combustion in the CI engine are shown in Fig. 10.63.

1. **First stage—ignition delay period:** During the first stage, some fuel has been admitted but has not yet been ignited. The ignition delay is counted from the start of injection to the point where the pressure-time (crankshaft rotation) curve separates from the pure air compression curve. The delay period is a sort of preparatory phase.

2. **Second stage—rapid or uncontrolled combustion (premixed flame):** In the second stage, the pressure rise is rapid because during the delay period the fuel droplets have had time to speed over a wide area and they have fresh air all around them. This period is counted from the end of the delay period to the point of maximum pressure on the indicator diagram. About one-third of the heat is evolved during this period.

Figure 10.63 *Stages of combustion in a CI engine*

3. **Third stage—controlled combustion (diffusion flame):** At the end of the second stage, the temperature and the pressure are so high that the fuel droplets injected during the last stage burn almost as they enter and any further pressure rise can be controlled by the injection rate. The period of controlled combustion is assumed to end at maximum cycle temperature. About 70%–80% of total heat of the fuel is supplied during this period.
4. **Fourth stage—after burning:** Due to poor distribution of fuel particles, combustion continues during the part of the remainder of the expansion stroke. This after-burning can be called the fourth stage of combustion. The after-burning period is about 70°–80° of crank angle from the TDC. The total heat evolved during the end of entire combustion process is 95–97% and 3–5% of heat goes to unburnt fuel.

10.18.2 Delay Period or Ignition Delay

In Fig. 10.64, the delay period is shown on pressure-crank angle (or time) diagram between points *a* and *b*. Point *a* represents the point of injection and point *b* represents the time at which the pressure curve (caused by combustion)

first separates from the compression curve (non-firing or motoring curve). This ignition delay period can be roughly divided into two parts.

1. **Physical delay:** The period of physical delay is the time between the beginning of injection and the attainment of chemical reaction conditions. In the physical delay period, the fuel is atomised, vapourised, mixed with air, and raised in temperature.
2. **Chemical delay:** In this period, the pre-flame reactions start slowly and then accelerate until local ignition takes place. Generally, chemical delay is longer than the physical delay. The delay period refers to the sum of physical and chemical delay. The ignition lag in SI engine is basically equivalent to the chemical delay in CI engine. There is no component like physical delay in SI engine.

Figure 10.64 *Pressure-time diagram illustrating ignition delay*

10.18.3 Variables Affecting Delay Period

1. **Fuel:** A lower pre-ignition temperature means a wider margin between it and the temperature of the compressed air and hence delay period. A higher cetane number means a lower delay period and smoother engine operation. Other properties of fuel which affect delay period are: volatility, latent heat, viscosity, and surface tension.
2. **Injection pressure or size of droplet:** The smaller the size and greater the number of droplets, the larger will be aggregate area of inflammation and therefore the greater the uncontrolled pressure rise. The size of the droplets depends on the injection pressure. Therefore, lower the injection pressure the lower the rate of pressure rise during the uncontrolled phase and smoother the running.

3. **Injection advance angle:** The delay period increases with increase in injection advance angle because the pressures and temperatures are lower when the injection begins. When the injection advance angles are small, the delay period reduces and operation of the engine is smoother but the power is reduced because larger amount of fuel burns during after-burning. The optimum angle of injection advance is 12°–20° before TDC.
4. **Compression ratio:** Increase in compression ratio reduces the delay period as it raises both temperature and density. However, higher compression ratio decreases the volumetric efficiency and power. It also decreases mechanical efficiency. In practice, the engine designer uses the lowest compression ratio which would satisfy the needs of easy cold starting and light load running at high speeds.
5. **Intake temperature:** Increasing the intake temperature would result in increase in the compressed air temperature, which would reduce the delay period. However, it would reduce the density of air and hence volumetric efficiency and power output.
6. **Jacket water temperature:** Delay period is reduced with increase in jacket water temperature or compressed air temperature increases.
7. **Fuel temperature:** Increase in fuel temperature would reduce both physical and chemical delay periods.
8. **Intake pressure or supercharging:** Supercharging reduces the auto-ignition temperature and hence reduces delay period.
9. **Speed:** As the engine speed increases, the loss of heat during compression decreases with the result that both the temperature and pressure of the compressed air tend to rise, thus reducing the delay period.

Table 10.9 *Effect of variables on delay period in CI engine*

1. 10. **Air-fuel ratio (load):** With increase in air-fuel ratio the combustion temperatures are lowered and hence the delay period increases. With increase in load, air-fuel ratio decreases operating temperature increase and hence, delay period decreases.

2. 11. **Engine size:** The engine size has little effect on the delay period in milliseconds but crank angle decreases.
3. 12. **Type of combustion chamber:** A pre-combustion chamber gives shorter delay period as compared to an open type of combustion chamber.

The summary of the effect of delay period in a CI engine is given in Table 10.9.

10.19 ❐ KNOCKING IN CI ENGINES

The fuel injection takes place over a definite period. Consequently, the first droplets injected are passing through the ignition delay, additional droplets are being injected into the chamber. If the ignition delay of the injected fuel is small the first droplets start burning in a relatively short time after injection and relatively small amount of fuel will be accumulated in the chamber. Therefore, the rate of pressure rise will be smooth

as shown in Fig. 10.65(a). However, if the delay period is longer, the accumulation of fuel will be larger as the actual burning of the first droplets is delayed. When actual burning starts, the accumulated fuel and additional fuel injected starts burning and can cause too rapid rate of pressure rise. This rapid rate of pressure rise causes pulsating combustion and creates heavy noise. This pulsating variation of pressure due to sudden burning of accumulated fuel is shown in Fig. 10.65(b). This type of abnormal combustion is known as *knocking*. This knocking occurs during the initial period over the delay period when the burning of the first portion of the fuel is uncontrolled, thus giving rise to an excessive rate of pressure rise.

Figure 10.65 *Combustion in CI engine: (a) Without knocking, (b) With knocking*

10.19.1 Factors Affecting Knocking in CI Engines

The factors which influence the delay period also influence the knocking tendency of CI engines are as follows.

1. If the injection of fuel is too far advanced, the rate of pressure rise during auto-ignition is very high and cause knocking.
2. Inferior fuels (having lower cetane number) promote diesel knock but this can be avoided by using better types of fuel (higher cetane number).
3. Fuel injection parameters (better penetration and distribution) and better combustion chamber design (split type) also influence knocking tendency of the engine.
4. Initial condition of the air (higher temperature or higher pressure in case of supercharged engine or both) influences knocking tendency. Higher pressure and temperature reduces knocking tendency.
5. The fuel having longer *delay period* and higher *self-ignition* temperature leads to knocking.

With proper design of combustion chamber and injecting the fuel uniformly over a hot surface of the combustion chamber, the detonation can be avoided.

10.19.2 Controlling the Knocking

If the delay period is long, a large amount of fuel will be injected and accumulated in the chamber. The auto-ignition of this large amount of fuel may cause high rate of pressure rise and high maximum pressure which may cause knocking in diesel engines. The methods for controlling diesel knock are as follows:

1. **Cetane number of the fuel used:** Cetane number is a scale for comparing the ignition delay period of various diesel fuels. A higher Cetane number means a lower delay period, and smoother engine operation, because it reduces the self-ignition temperature. Some compounds called additives or dopes are used to improve combustion performance of fuels. Various additives used for diesel fuels are: Isopropyl nitrate, Amyl nitrate, Heptyl peroxide and Butyl peroxide. These dopes delay the auto-ignition but they increase NO_x drastically.

Figure 10.66 *Required rate of fuel supply to avoid detonation*

2. **Compression ratio:** Increase in compression ratio reduces the delay period as it raises both temperature and density. At the same time the minimum auto-ignition temperature decreases which reduces the time of reaction when fuel is injected. Thus the delay period decreases. With increase of compression ratio the unused air would be much more decreasing the volumetric efficiency and power. It further decreases the mechanical efficiency due to increase in weight of reciprocating parts.

3. **Air-fuel ratio:** With decrease in air-fuel ratio (richer mixture) the combustion temperatures are increased and cylinder wall temperatures are increased and hence the delay period decreases. With increase of load air-fuel ratio decreases, operating temperature increases and hence delay period decreases.
4. **Injection pressure:** Lower the injection pressure, the lower the rate of pressure rise during the uncontrolled phase and smoother the running of engine. Lower injection pressure gives rise to fuel droplets of smaller size to obtain largest surface-volume ratio resulting in less physical delay. Large droplets of fuel results in too low subsequent burning.
5. **Controlling the rate of fuel supply:** It can be reduced by controlling the initial rate of fuel delivery without affecting efficient combustion condition. A small amount of fuel should be supplied until combustion starts and more fuel should be supplied as the inside combustion chamber condition is suitable (because of high temperature inside) to burn more fuel. In this case, the cam shape is so designed that it gives a low initial injection followed by a main injection at a higher rate as shown in Fig 10.66.
6. **Knock reducing fuel injector:** This type of injector is developed so that it can avoid the sudden increase in pressure inside the combustion chamber due to the accumulated fuel. In this case, the injection pressure of 100 bar is used with a semi-flexible needle valve.

10.19.3 Comparison of Knocking in SI and CI Engines

It is interesting to compare the knocking phenomenon in SI and CI engines. This phenomenon in both engines is fundamentally similar as it involves *auto-ignition,* subject to the ignition-lag

characteristics of A:F mixture supplied to the engine.

1. In SI engines, the knocking occurs near the end of combustion, whereas in CI engine, this occurs at the beginning of combustion as shown in Fig. 10.67.

Figure 10.67 *Knocking in SI and CI engines*

2. The knocking in SI engine takes place in a homogeneous mixture; therefore, the rate of pressure rise and maximum pressure is considerably high. In case of a CI engine, the mixture is not homogeneous and the rate of pressure is lower than in an SI engine.
3. The question of pre-ignition does not arise in CI engines as the fuel is supplied only near the end of compression stroke.
4. It is easier to distinguish between knocking and non-knocking operations in SI engines as the human ear easily finds the difference. However, in a CI engine, the normal ignition is by auto-ignition and the rate of pressure-rise under normal operation is considerably higher (10 bar against 2.5 bar for SI engines) and causes high noise. The noise level becomes excessive under knocking condition. Therefore, there is no definite distinction between *normal* and *knocking* combustion.

The knocking in SI engines is due to the auto-ignition of the last part of the charge. To avoid this, the fuel must have a *long-delay period* and high self-ignition temperature.

To avoid knock in the CI engine, the

delay period should be as small as possible and the fuel self-ignition temperature should be as low as possible.

The factors which reduce knock in SI and CI engines are given in Table 10.10.

Due to this dissimilarity in the time of starting of the knock in SI and CI engines, *the conditions which reduce the knock tendency in the SI engine increase the knock tendency in CI engines*. The fuels which are better from the point of view of avoiding detonation in SI engines may promote detonation in CI engines. Therefore, good SI fuels are poor CI fuels. In terms of fuel rating, diesel oil has high cetane number (40–60) and low octane number (30) and

petrol has high octane number (80–90) and low cetane number (20).

Table 10.10 *Factors which reduce knock in SI and CI engines*

10.20 ❐ COMBUSTION CHAMBERS FOR CI ENGINES

The most important function of the CI engine combustion chamber is to provide proper mixing of fuel and air in a short time. For this purpose, an organised air movement, called air swirl, is provided to produce high relative velocity between the fuel droplets and air.

There are three basic methods of generating swirl in a CI engine combustion chamber as follows.

1. By directing the flow of the air during its entry to the cylinder, known as induction swirl. This method is used in open combustion chambers shown in Figs 10.68 (a) to (d).
2. By enforcing air through a tangential passage into a separate swirl chamber during the compression stroke, known as compression swirl. This method is used in swirl chambers. This type of chamber is shown in Fig. 10.69.

Figure 10.68 *Open combustion chambers: (a) Shallow depth chamber, (b) Hemispherical chamber, (c) Cylinderical chamber, (d) Toroidal chamber*

Figure 10.69 *Ricardo swirl chamber comet, mark II*

Figure 10.70 *Combustion-induced swirl chambers: (a) Precombustion chamber, (b) Lanova air-cell combustion chamber*

1. 3. By use of the initial pressure rise due to partial combustion to create swirl turbulence, known as combustion induced swirl. This method is used in pre-combustion chambers and air-cell chambers. These type of chambers are shown in Figs 10.70(a) and (b).

The comparison of induction and compression swirl is given in Table 10.11 and comparison of combustion characteristics in Table 10.12.

Table 10.11 *Comparison of induction and compression swirl*

Table 10.12 *Comparison of combustion chamber characteristics (Mainly for four-stroke non-supercharged high-speed engines)*

10.21 ❐ LUBRICATION SYSTEMS

All moving parts of an internal combustion engine have relative motion and rub against each other. The lubrication is required to reduce this rubbing action and increases the life of engine. The purpose of lubrication is to reduce the rubbing action between different machine parts having relative motion and to remove the heat generated inside the cylinder. Due to friction between rubbing parts and combustion of fuel in the cylinder, the temperature of various parts, cylinder, and piston may rise enormously if the engine is not

cooled. The lubrication system will also be seriously affected, leading to damage to the cylinder and piston. Therefore, it is essential to provide lubrication and cooling to various engine parts.

10.21.1 Functions of a Lubricating System

The important functions of a lubricating system are as follows:

1. **Lubrication:** The main function of the lubricating system is to keep the moving parts sliding freely past each other and thus, reduce the engine friction and wear.
2. **Cooling:** To keep the surfaces cool by taking away a part of their heat through the oil passing over them. This requires oil to possess good oxidation stability.
3. **Cleaning:** To keep the bearings and piston rings clean of the products of wear and combustion by washing them away.
4. **Sealing:** The lubricating oil must form a good seal between piston rings and cylinder walls. Therefore, it must possess adequate viscosity stability.
5. **Reduction of noise:** Lubrication reduces the noise of the engine and other sliding parts.

10.21.2 Desirable Properties of a Lubricating Oil

The desirable properties of a lubricant are as follows:

1. It should be available in a wide range of viscosities.
2. There should be little change in the viscosity of the oil with a change in temperature.
3. The oil should be chemically stable at all temperatures encountered in its application.
4. The oil should have sufficient specific heat to carry away frictional heat without abnormal rise in temperature.
5. It should be commercially available at a reasonable cost.

10.21.3 Lubricating Systems Types

There are two types of lubricating systems:

1. **Splash lubricating system:** The arrangement of a splash lubricating system is shown in Fig. 10.71. This method is generally used for a vertical engine with a closed crankcase. The sump is located at the bottom of the crankcase. When the engine crankshaft rotates, the big end of the connecting rod splashes oil by centrifugal action. The connecting rod big end has a hollow pipe called a scoop which is fitted to the bearing cap and pointed towards the direction of rotation of the crankshaft. The lubricating oil passing through the scoop lubricate the big end bearing and gudgeon pin bearing. All other parts are lubricated by the splash. Excess oil is collected in the troughs, is provided with overflows, and collected in the main sump. The level of the oil in the trough is maintained constant. The dripping from the cylinders is also collected in the sump. The oil from the sump is recirculated with the help of a pump.

   The limitations of this system are as follows:

   1. Inability to regulate the quantity of oil splashed against the cylinder wall.
   2. Inability to keep the oil from getting past the piston head into the combustion chamber, burning with the fuel and passing out with exhaust gases.

2. **Pressure feed lubricating system:** This system is shown in Fig. 10.72. Such a system supplies oil under pressure directly to the connecting rod bearing, crankshaft bearings, valve gear, and the camshaft drive. Indirect supplies reach the cylinder walls, gudgeon pin, the distributor, and pump drives.

   Oil is carried in the sump and circulated by the gear pump which sucks from the sump through a strainer. The pump delivery pressure is controlled by a relief valve and the oil passes through a very fine filter before it reaches the main distributor gallery from the various bearings, surfaces, and gears.

   After lubricating the big end bearings, the oil is fed to the gudgeon pins through the oilway in the connecting rod and further squirted into the cylinder wall.

3. **Charge lubricating system:** It does not require oil-filter and oil sump. In this system, the lubricating oil is pre-mixed with the fuel which carries the lubricating oil in the cylinder to lubricate the piston and cylinder. Most of the oil burns with the fuel and is carried away with the exhaust gases. The lubricating oil cannot be recovered in this system. The system is generally used for two stroke spark ignition engines of scooters and motorcycles.

Figure 10.71 *Splash lubrication system*

Figure 10.72 W*et sump high pressure lubrication system*

The main disadvantages of this system are as follows:

1. Carbon deposits due to burning of oil on the spark plug
2. Non-recovery of the oil used

The advantages of this system are as follows:

1. It does not require a separate lubricating system and is the most economical one.
2. There is no risk of failure of lubrication system.
3. The lubricating oil supplied is regulated at various loads and speeds by the increased fuel flow.

10.21.4 Lubricating System for IC Engines

Various lubricating systems used for internal combustion engines may be classified as follows:

1. **Mist lubrication system:** This system is used for two-stroke cycle engines. Most of these engines are crank-charged, that is, they employ crankcase compression and thus, are not suitable for crankcase lubrication.

   Such engines are lubricated by adding 2–3% lubricating oil in the fuel tank. The oil and the fuel mixture is inducted through the carburettor. The gasoline is vapourised; and the oil, in its form of mist, goes through a crankcase into the cylinder. The oil which impinges on the crankcase walls lubricates the main and connecting and bearing, the rest of the oil which passes on to the cylinder during charging and scavenging periods, lubricates the piston, piston rings, and the cylinder.

The two-stroke engine is very sensitive to a particular oil and fuel combination. The composition of fuels and lubricants used influence the exhaust smoke, internal corrosion, bearing life, ring and cylinder bore wear, ring sticking, exhaust and combustion chamber deposits, and one of the most irritating and difficult problem of spark plug fouling and whishering. Therefore, specially formulated ashless oils are used for two-stroke engines.

The fuel/oil ratio used is also important for good performance. A fuel/oil ratio of 40 to 50:1 is optimum. Higher ratios increase the rate of wear and lower ratios result in spark plug fouling.

The main advantage of this system is simplicity and low cost because no oil pump, filter, etc., are required. However, this simplicity is at the cost of many troubles some of which are enumerated as follows:

1. Some of the lubricating oil remains invariably in combustion chamber. This heavy oil when burned, and still worse, when partially burned in the combustion chamber, leads to heavy exhaust emissions and formation of heavy deposits on the piston crown, ring grooves, and the exhaust port which interferes with efficient engine operation.
2. One of the main functions of the lubricating oil is protection of anti-friction bearings, etc., against corrosion. Since the oil comes in close contact with acidic vapours produced during the combustion process, it rapidly loses its anti-corrosion properties

resulting in corrosion damage of bearings.

3. For effective lubrication, oil and the fuel must thoroughly mixed. This requires either separate mixing prior to use or use of some additive to give the oil good mixing characteristics.
4. One important limitation of this system is oil starvation of the working parts, especially when the throttle is closed on a descent on a long hill. A closed throttle means no fuel, and, hence, no oil. The prolonged absence of oil so produced may result in overheating and piston seizure. This oil starvation can be controlled if the driver periodically releases the throttle to replenish for the complete absence of oil while descending a hill.
5. Due to high exhaust temperature and less efficient scavenging, the crankcase oil is diluted. In addition, some lubricating oil also burns in the combustion chamber. This results in about 5–15% high lubrication consumption for two-stroke engines as compared to four-stroke engines of similar size.
6. Since there is no control over the lubricating oil, once introduced with fuel, most of the two-stroke engines are over-oiled most of the time.

Some manufacturers use a separate oil injection pump to inject the oil directly into the carburettor and the amount of the oil is regulated according to engine load and speed. This system completely avoids the oils starvation problem discussed above. The oil consumption is also reduced. This results in less deposits and less spark plug fouling problems. Since in this system the main bearings are excluded from the crankcase and instead receive lubricating oil from a separate pump, the corrosion damage of bearings is also eliminated.

2. **Wet sump lubricating system:** The wet sump lubricating

system is shown in Fig. 10.73. In this system, the sump is always full of oil. Oil is drawn from the sump by an oil pump through an oil strainer. A pressure relief valve is provided which automatically maintains the delivery pressure constant. If the pressure exceeds the predetermined pressure, the valve opens and allows some of the oil to return to the sump and relieves the oil pressure in the system. The oil from the pump goes to the bearings and part of it passes through the filter which removes solid particles from the oil. As all the oil is not passed through the filter, the system is known as by-pass filtering system. The advantage of this system is that a clogged filter will not restrict the flow of oil to the engine.

3. **Dry sump lubricating system:** In the dry sump lubricating system as shown in Fig. 10.74, the sump does not contain oil and all the oil required for lubrication remains in circulation. High speed racing cars and military jeeps use this type of system.

An auxiliary tank is used to supply the oil to the main bearings with the help of the pump. The oil returns to the tray and is then returned to the auxiliary tank by the scavenging pump. If the filter gets clogged, the pressure relief valve opens, permitting oil to flow by passing the filter and reaches the supply tank. The oil is then recirculated to the bearings from the supply tank. A separate oil cooler is used to cool the oil.

10.21.5 Lubrication of Different Engine Parts

1. **Lubrication of main bearings:** The main bearings are lubricated satisfactorily with the help of a ring or a chain-type feeder. The arrangement of the system is shown in Fig. 10.75.

The oil ring rests on the main shaft where a small portion of the main bearing shell has been cut away as shown in the figure. The lower end of

the oil ring is allowed to submerge in the oil bath as shown in Fig. 10.73. The oil ring rotates with the main shaft and carries the oil from the oil bath to the bearing and is distributed to the bearing through the oil groove. The surplus oil flows to the ends of the bearing and drops back into the oil bath. Chains or more rings are provided instead of single ring to carry more oil.

Figure 10.73 *Wet sump lubricating system*

Figure 10.74 *Dry sump lubricating system*

Figure 10.75 *Ring lubrication*

This type of lubrication is more useful for medium speed engines because at high speeds the oil will be thrown off due to centrifugal force but at low speeds, the amount of oil carried is not sufficient.

2. **Lubrication of cylinder and small end bearing of connecting rod:** The cylinder and a small end bearing (gudgeon pin) are lubricated with the help of side feed lubricator as shown in Fig. 10.76.

The oil is supplied to the surface of piston through the hole provided in the cylinder wall. The oil falling on the surface of the piston is spread over the surface of the cylinder walls.

3. **Lubrication of crank and gudgeon pin:** The scoops are provided to the caps of big-end-bearing of the connecting rods as shown in Fig. 10.76. These scoops dip into the splash toughs when the pistons are at BDC and splashes the oil in all

directions with the rotation of cranks. A thin mist of splashed oil settles on the surface of the cylinder and on the other parts which are to be lubricated. The main bearings and cam shaft bearings are provided with small pockets and the oil splashed by scoops is collected in those pockets and is used for lubrication. The surplus oil falls back into the oil sump.

Figure 10.76 *Side feed lubricator*

Many times, a through hole is provided in the connecting rod and it is connected with the scoop; therefore, some oil is supplied to the crank pin and through the hole to the gudgeon pin. The oil level in the splash tough is maintained with the help of oil pump. The arrangement of scoop with connecting rod is shown in Fig. 10.77.

4. **Lubrication of big end (crank pin) and small end (Gudgeon pin) of connecting rod:** The crank pin and gudgeon pin are lubricated by ring as shown in Fig. 10.78.

The arrangement of the system used is shown in Fig. 10.78. In this system, the oil is supplied to the ring from the oil feed pump which is fitted concentric on the crank shaft. The centrifugal force is given to the oil as the oil ring rotates with the crank shaft and the lubricating

oil is forced to the crank pin and to the gudgeon pin through the holes provided in the crank pin and connecting rod as shown in Fig. 10.77. This system of lubrication is known as ring lubrication.

Figure 10.77 *Scoope used in splash lubrication*

Figure 10.78 *Lubrication system for big end of connecting rod*

10.22 ❐ NECESSITY OF IC ENGINE COOLING

The cooling is provided to avoid the effects of overheating as follows:

1. The high temperature reduces the strength of the piston and piston rings and uneven expansion of cylinder and piston may cause the seizure of the piston.
2. The high temperature may cause the decomposition of the lubricating oil and lubrication between the cylinder wall and piston may break down, resulting in a scuffing of the piston.
3. If the temperature around the valve exceeds 250°C, the overheating of the valves may cause the scuff of the valve guides, due to lubrication break down. Further increase in temperature may cause the burning of valves and valve seats.
4. The tendency of the detonation increases with an increase in temperature of the cylinder body.
5. The pre-ignition of the charge is possible in spark ignition engines if the ignition parts initially are at high temperature.

10.22.1 Types of Cooling Systems

There are two types of cooling system as per the coolant used—air cooling and water cooling system.

Air Cooling System

In this system, air is used as a cooling medium and it is used for small capacity engines. Earlier, it was used for big capacity air-craft engines as water cooling was not practically possible as the weight of water cooling system is very high compared with air-cooling system.

As mentioned earlier, the heat transfer coefficient for air-cooling is very low. This heat transfer coefficient can be increased further by using the forced flow of air over the engine surface as done in aero-engines. The heat transfer

coefficient with air-cooling with forced circulation is also considerably lower (50 W/m^2-K) when compared with water cooling system. The other method of increasing the heat transfer rate from the cylinder's surface is to increase the surface area by providing the fins. The use of fins increases the heat transfer surface area by 5 to 10 times of its original value. The air-cooled systems, the forced circulation of air with increased surface area by using the fins, is commonly used in practice for aero-engines and motor cycle engines.

The sectional view of an engine-cylinder with fins is shown in Fig. 10.79. More fins are used near the exhaust valve and cylinder head where

the possibility of occurrence of maximum temperature exists.

The cooling fins are either cast integral with the cylinder and cylinder head or they are fixed to the cylinder block separately. The number of the fins used are two to three in case of cast fins and four to five in case of machined fins per centimetre. The height of the fin depends on the type of material and the manufacturing process used. Generally, the height of the fin used lies between 2 cm and 5 cm and the fin spacing is limited to 2.5 mm.

Advantages of Air Cooling

1. Water jacket, radiator and water pump are not required. This reduces the size of the engine and its weight.
2. Simpler engine design.
3. Less sensitive to climatic conditions engine.
4. Due to small thermal losses, the specific fuel consumption is low.

5. Better warm-up performance results in low wear of cylinder.
6. Sustained engine performance due to reduced carbon deposit on combustion chamber.
7. Easier control of cooling systems compared to water-cooled engines.
8. An air-cooled engine can take up some degree of damage.

Disadvantages of Air Cooling

1. Air fan creates noise.
2. The volumetric efficiency is lower due to higher cylinder head temperature.
3. High specified output engines cannot be air-cooled due to the complex nature of the fins that are required.
4. Air cooling results in higher engine temperature. This necessitates the provision of bigger clearances between the various parts of the engine.

Figure 10.79 *Air cooling of engine cylinder*

Water Cooling System

The water cooling system is further subdivided into two groups:

1. **Thermo-siphon system:** The arrangement of the system is shown in Fig. 10.80. The force (pressure head) required to circulate the water through the system is the difference in pressure head due to hot and cold water. This force is given by

$$F = h(\omega c - \omega h) = h \times \Delta\omega$$

where h = Height of the radiator tubes through which the water is circulated.

ω_c = Weight density of cold water.

ω_h = Weight density of hot water.

The difference in density is limited as the rise in temperature of the water passing through the engine is limited. The rate of circulation is less as the force causing the flow of water is limited. This cannot be used for heavy duty engines as the heat carrying capacity of this system is limited. The water as passed through the radiator is cooled by the flow of air passed over the radiator tubes by the cooling fans. The cooled water rises to the cylinder jacket; takes the heat from the cylinder wall and then it enters into the radiator from the top header and comes down. It is cooled as it is passed through the radiator tubes.

The limitations of this system are as follows:

1. The engine should be placed as low as possible in relation to the radiator as the force causing the flow is limited by the temperature difference of hot and cold water.
2. The water level in the system should not fall below the level of the delivery pipe; otherwise, the circulation of water in the system will stop. This causes the boiling of water and formation of steam, resulting in further loss of water which may damage

the engine in a short period.
3. The use of this system is recommended for small capacity engines.
4. The main drawback of this system is that cooling depends only on the temperature and is independent of engine speed. The rate of circulation is slow.

Due to these limitations, this system is rarely used at present.

Figure 10.80 *Thermo iphon system*

Figure 10.81 *Forced pump system*

2. **Forced pump system:** In a thermo-siphon system, the height of the radiator should be kept above the engine as the flow of water takes place by natural circulation.

To avoid this limitation, a pump is incorporated in the system for water circulation. The arrangement of the system is shown in Fig. 10.81. The pump is driven from the engine shaft with the help of the belt.

The major advantage of this system is effective and positive cooling of all the parts of the engine. It can easily take the overload as the engine speed increases, water circulation is also increased and same effective cooling can be maintained by this system.

Advantages of Forced Circulation System

1. The increased rate of coolant flow improves the heat transfer between the gases and water. This reduces the quantity of water to be circulated and thereby reduces both the bulk and

weight of the engine because smaller coolant passages can be used.

2. The high temperature areas such as spark plug; exhaust valve seats and exhaust ports can be effectively cooled and reduces local overheating.
3. The formation of steam pockets in critical areas that restrict the coolant circulation and hence the heat transfer, can be avoided by maintaining the pressure in the system.
4. Increased coolant velocities allow a higher mean temperature difference between the water and the air passing through radiator matrix. This reduces the size of the radiator.
5. When the vehicle is operated in mountainous regions, loss of water through boiling may occur in thermo-siphon system is used because the boiling temperature of water decreases with the decrease in atmospheric pressure at high altitudes.

Disadvantages of Forced Circulation System

1. There is a possibility of overcooling as the cooling is independent of temperature.
2. There is a possibility of overheating the engine when the automobile is climbing a hill where the fuel burning rate is increased and water flow rate is reduced as part of the head is lost to overcome the inclination of the road.
3. As soon as engine stops, the cooling also stops. This is undesirable because the cooling should continue for some time till the temperatures are brought to normal values.

Cooling With Thermostatic Regulator

An important feature of the cooling system of present-day cars is the inclusion of a thermostat. This is generally fitted in the upper radiator hose and its purpose is to prevent the

circulation of water through radiator until the cylinder block and head have reached a temperature of about 75°C. If the water circulation starts immediately as the engine has been started (as in the absence of thermostat), the engine cannot run efficiently when all the water in the cooling system has been raised to a temperature near the boiling point (90°C). By preventing circulation before the engine is warm, the engine gets warm much more quickly. In consequence, there is less danger of the oil being washed away from the cylinder walls by wet petrol drawn into the cylinders while the engine is cold. Less petrol is wasted while the engine warms up and the car can be driven more quickly after the engine has been

started. The arrangement of this system is shown in Fig. 10.82.

The thermostat consists of a valve attached to a bellows containing volatile liquid such as ether. Heating of bellow by the water around them causes vapourisation of the liquid, expansion of the bellows, and therefore, opening of the valve.

When the forced circulation is used, it is necessary to use a bypass pipe from the engine side of the thermostat (as shown in Fig. 10.80) to the lower water hose so that the circulation pressure of the water cannot force to open the valve before the correct temperature has been reached.

Figure 10.82 *Thermo-syphon cooling system for IC engines*

Evaporative Cooling System

This is not preferred until there is acute shortage of cooling water. The arrangement is shown in Fig. 10.83. In this system, the water coming out of engine jacket is allowed to be heated to 100°C. This method of cooling utilises the latent heat of water to obtain cooling with minimum water. The cooling circuit is such that the coolant is always maintained in liquid form but the steam formed is flashed off in a separate vessel as shown in Fig. 10.83. This system is generally used for big-capacity stationary engines used for power generation (2 MW).

Advantages of Water Cooled Engines

1. The engine design is compact due to high heat transfer

coefficient of water.
2. Due to high latent heat of water, overheating troubles are eliminated.
3. The engine can be installed anywhere in the vehicle.
4. The volumetric efficiency is higher than air-cooled engine.

Disadvantages of Water-cooled Engines

1. The need for a radiator and a pump increases the weight and dimensions of the engine.
2. It requires more maintenance.
3. The engine performance is more sensitive to climatic conditions.
4. The warm-up performance is poor resulting in greater cylinder wear.
5. Pump absorbs slightly higher power than fan.

Open and Closed System

In open this system, the hot water coming out from the engine jacket is not circulated through the pump directly but it is cooled either in cooling tower or spray pond as shown in Fig. 10.84.

Figure 10.83 *Evaporative cooling system*

Figure 10.84 *Open cooling system*

With the help of a spray, the water is cooled by partly evaporating the water

(1%) and absorbing the latent heat required from the remaining water. This evaporated water is carried by the air. The effective cooling of water depends upon the relative humidity of the surrounding area. This system is generally used where more than adequate water is available.

In closed system, the hot water coming out from the engine is cooled in the radiator, heat exchanger, or cooling tower. In this system, the same water is used repeatedly without exposing to the air directly as in the case of an open system.

Impurities in water and the consequent danger of scale formation are major problems in an open system. Due to

water evaporation, which does not carry impurities, the concentration of salt in the water after every pass increases and danger of scaling also increases. In closed system, the hot water cooling methods are shown in Figs 10.85(a) to (c).

In a closed system, the same water remains in the system and recooled repeatedly. If it is pure at the start, it remains pure throughout the period of use and there is no danger of scaling.

Figure 10.85 *Closed cooling systems: (a) Closed system (radiator type), (b) Closed system (heat exchanger) using raw water, (c) Closed system using cooling tower*

Advanced Cooling System

There are two main disadvantages with the conventional cooling systems. First,

large volume of coolant in the primary circuit can lead to a slow engine warm up and second, the cooling system tends to overcool the engine parts when the engine is running at part load conditions. To overcome these difficulties, advanced cooling systems are used.

The main purpose of these systems is to maintain more uniform temperature and less sensitive to the load and speed of the engine. The friction loss between the piston and the cylinder is reduced if the liner temperature is raised. The significance of reduction in friction loss increases as load is reduced because this leads to reduction in fuel consumption. A rise in liner temperature may increase

the emission of NO_x. This can be reduced either by retarding the ignition timing or by using exhaust gas recirculation (EGR). Both these remedies impose a fuel consumption penalty and the fuel saving gained by reducing the frictional loss. The NO_x emission is problematic in diesel engine at full load conditions.

With SI engines, another problem that can occur at full load is the reduction in the knock margin caused by higher cylinder temperatures. This also can be remedied by EGR or retarding the ignition timing.

A few methods used for this purpose are described in brief.

It is developed for cooling diesel engines and to provide cooling only where it is needed; at a rate proportional to the local heat flux.

A precision-cooled system has small local passages which are used to cool critical regions such as injector nozzle, exhaust valve region, and valve guides, with much space in the head filled with air.

With precision cooling, many regions are not directly cooled and their temperatures rise, but the system is designed to keep the temperature within safe limits. The corresponding benefit is a more even temperature distribution, producing less thermal strain and

possible elimination of hot spots to allow higher compression ratios in SI engine.

10.22.3 Dual Circuit Cooling

In this process of cooling, there are two separate cooling circuits—one for the head and the other for the cylinder block. The higher block temperature reduces friction losses and reduces fuel consumption, particularly at part loads. The lower coolant temperature in the cylinder head tends to reduce the indicated thermal efficiency but the risk of knock is reduced in SI engines and higher compression ratio can be used. The overall effect is an improved efficiency.

Increasing the coolant temperature

passing through cylinder block reduces the fuel consumption and hydrocarbon emissions but increases NO_x emissions.

10.22.4 Disadvantages of Overcooling

Should we cool the engine as much as we can? No. The engine must never be overcooled. The engine must always be kept sufficiently hot to ensure smooth and efficient operation. Extremely low engine temperatures may be difficult to start and above all, the low temperature corrosion assumes such a significant magnitude (see Fig. 10.86) that the engine life is greatly reduced. At low temperatures, the sulphurous and sulphuric acids resulting from combustion of fuel (fuel always contains some sulphur) attack the cylinder barrel.

The dew points of these acids vary with pressure, and hence, the critical temperature, at which corrosion assumes significant properties, varies along the cylinder barrel. To avoid condensation of acids the coolant temperature should be greater than 70°C. Thus, the cooling system should not only cool but must also keep the cylinder liner temperature above a minimum level to avoid corrosion and ensure good warm up performance of the engine.

Figure 10.86 *Effect of temperature on corrosion*

10.23 ❐ ENGINE RADIATORS

The function of the radiator is to cool hot water coming out of the engine with the help of air. In the radiator, the heat transfer from hot water to the air takes

place by conduction and convection.

The cooling effect in the radiator is enhanced by providing fins over the surface of tubes carrying the hot water to be cooled. The fins are also so arranged that some turbulence is generated in air passages which further help to increase heat transfer coefficient to the air side.

A commonly used cross-flow type radiator is shown in Fig. 10.87. The fins are not shown.

10.23.1 Radiator Matrix

Figure 10.87 shows four different types of matrices (fins) commonly used in practice.

In the tube and fin type shown in Fig. 10.87(a), a series of long tubes extending from top to bottom of the radiator are surrounded by simple straight metallic fins. The matrix has greater structural strength.

The improvement over the tube and fin type is the tube and corrugated fin type matrix as shown in Fig. 10.87(b). The water tubes are made of an oval-shape section and zig-zag copper ribbons are used to provide secondary heat transfer areas and air turbulence. This combines the good heat transfer characteristics of film type matrix and structural strength of the tube and fin type matrix.

Figure 10.87(c) shows the film-type matrix which is also known as the

ribbon-cellular matrix. It consists of a pair of thin metal ribbons soldered together along their edges so as to form a water-way running from header tank to the bottom tank of the radiator. The zig-zig copper ribbons forms an air passage. In this case, the metal surface area is significantly increased and higher turbulence is also created.

Figure 10.87 *Different types of radiator matrices: (a) Plate type matrix, (b) Ribbon cellular type matrix, (c) Corrugated fin matrix, (d) Honey comb matrix*

Earlier, radiators used honey-comb matrix as shown in Fig. 10.87(d). The hexagons were packed in contact and bound by solder. This provided a continuous water passage between the circular parts of the tube. The cooling air was passed through the circular

tubes. This type of matrix is rarely used in present types of radiators.

In all the aforementioned matrices, the air flow was in cross-flow (perpendicular to the paper).

The materials used for matrices must have the following properties:

1. It must have good conductivity (Cast iron = 60 W/m-K, Aluminium = 210 W/m-K and Copper = 400 W/m-K).
2. The material must have good corrosion resistance.
3. It must possess the required strength.
4. It must be easily formable.
5. It must be easily available at a reasonable cost.

Yellow brass and copper meet all these requirements and are widely used.

10.23.2 Water Requirements of Radiator

The quantity of cooling water (Q) required to be circulated is given by

Figure 10.88 *Variation of amount of air required with power for air-cooled radiator*

where (ΔT) is permissible rise in temperature and the value of K depends upon specific fuel consumption and compression ratio.

The cooling water carries 4000 kJ/kWh for large engines and 550 kJ/kWh for small engines. If the rise in temperature is limited to 10–15°C, the outlet cooling water temperature is limited to 60°C for small or medium engine and 80°C for automobile engines.

Figure 10.88 shows the amount of air required for air-cooled radiator. It is clear from the graph, petrol engine requires much more air than diesel engine.

The power required for cooling the engine is given by

where A_f and ΔT_f are the fin area and temperature difference between the fin average surface temperature and air temperature, ρ_a is air density, and A_e is effective area.

The equation is developed with following assumptions.

1. A:F ratio supplied to engine is constant.
2. The temperature change in fin is assumed small.
3. The density and temperature of the cooling media are assumed to be unaffected by its flow through the radiator fin.

10.23.3 Fans

As mentioned earlier, an engine driven fan is mounted behind the radiator to increase the air flow over the radiator tubes. This increases the air-side heat

transfer coefficient and ultimately overall heat transfer coefficient.

A radiator fan is usually made of pressed steel blades, four or six in number depending upon the size of the radiator (capacity of engine). The power consumed by the fan depends upon the amount of air handled, diameter of the fan, its shape, and speed of the engine.

Figure 10.89 shows the effect of speed and diameter of fan on the quantity of air flow. Figure 10.90 shows the effect of fan diameter and air flow on the power consumed by the fan. For the given flow rate of air, the power required decreases slightly as the fan diameter increases, whereas for the same engine speed, the power required

increases with the fan diameter.

Figure 10.89 *Variation of diameter of fan on the quantity of air flow*

Figure 10.90 *Effect of fan diameter and air flow on the power consumed*

10.24 ❒ COOLING OF EXHAUST VALVE

In the liquid cooled engines, the spaces around the exhaust valve seating should be sufficiently large for the coolant to flow. Sometimes, the water circulation provides jets of water to be directed on the valve seating metal.

An efficient valve cooling requires methods of conducting the heat from the valve head to the cylinder, improved guides and metal, also to the lower part of the valve stem operating under cooler air conditions.

In order to enhance the conductivity of

the exhaust valve, the practice of making the valve stem and even the heads hollow is becoming popular. The hollow stem is filled with sodium which melts at 97°C and boils at 880°C. Thus, at valve operating temperatures, the valve is filled with conducting material. Such a valve is shown in Fig. 10.91 with a tapered plug seal driven into the hollow stem. The end of the stem is provided with a Stellite button, fused on the seat to prolong the wearing life of stem and end.

Figure 10.91 *Cooling of exhaust valve*

10.25 ❒ GOVERNING OF IC ENGINES

Governing is the process of varying the fuel supply to the engine in accordance with the load demand so that the engine

runs at practically constant speed. There are four methods of governing of IC engines.

1. **Hit and miss governing:** This method is used for small engines and is illustrated in Fig. 10.92. The rotational motion of the cam C actuates the rocker R through the roller D. The roller carries a pecker G which strikes against the pecker block H and lifts the valve V against the spring S. When the speed of the engine becomes excessive, the pecker block is lifted by the rod E as the sleeve of the governor rises up. The pecker is now unable to strike against and lift the valve off its seat and the engine performs an idle cycle because no fuel is now being supplied. This method of governing is quite simple but owing to the violent explosions which usually occur as a result of extra scavenging which take place immediately after a mixed explosion, produces uneven turning moment necessitating the use of heavy flywheel which increases the friction at the bearings and lowers the mechanical efficiency of the engine.

Figure 10.92 *Hit and miss governing*

2. **Qualitative governing:** In this method, a centrifugal governor is used to control the supply of fuel, whereas the supply of air remains constant. Thus the quality of air-fuel mixture is altered. This method is widely used in all heavy oil engines.
3. **Quantitative governing:** In this method, the quantity of air-fuel mixture flowing into the cylinder is varied. This may be done by decreasing the lift of the inlet valve or by throttling the mixture before it is made to enter the cylinder. This air-fuel ratio of the mixture is kept constant. This gives a more even turning moment and closer limits of speed variation. This method is widely used for governing petrol and large gas engines.
4. **Combination method:** This method uses qualitative and quantitative methods simultaneously.

10.26 ❐ RATING OF SI ENGINE FUELS-OCTANE NUMBER

The tendency to knock depends on the composition of fuel as mentioned earlier. Fuels differ widely in their ability to resist knock. The rating of a particular fuel is done by comparing its performance with that of a standard reference fuel which is usually a combination of iso-octane and n-heptane.

Iso-octane offers great resistance to knock and is arbitrarily assigned a rating of 100 octane number, *n*-heptane, a straight chain paraffin, on the other hand, detonates very rapidly therefore it is assigned a rating of zero-octane number. It is possible to match the knocking tendency of all commonly used fuels by blending these two

reference fuels in different proportions.

The percentage of *iso*-octane by volume in a mixture of *iso*-octane and *n*-heptane, which exactly matches the knocking intensity of a given fuel, in a standard engine under given standard operating conditions, is termed as ‘octane number’ of the fuel. An octane number of 80 means that the fuel has the same knocking tendency as a mixture of 80% *iso*-octane and 20% *n*-heptane by volume.

It has already been mentioned that the knocking tendency of an engine increases with increasing compression ratio. The higher the octane number of the fuel, the greater will be its resistance to knock and higher will be the

compression ratio which may be used without knocking. The power output and specific fuel consumption are functions of the compression ratio and therefore, these can be considered as a function of the octane number of the fuel. This fact indicates an importance of the octane number rating of fuels for SI engines. Higher octane number fuels increase the efficiency of the engine as high compression ratio can be used with higher octane number fuels.

The knocking tendency of the fuels used in aero-engines is more as they are supercharged engines; therefore, it is necessary to use the fuels having the highest octane number (largest resistance to knock). The fuels superior

in anti-knock qualities to *iso*-octane (having octane number greater than 100) are required. The addition of tetra-ethyl-lead (TEL) to gasoline produces marked an effect in reducing the knock tendency of the fuel. The addition of various amounts of one of these compounds (TEL is commonly used) to *iso*-octane produces fuels of greater anti-knock quality than *iso*-octane alone. The anti-knock quality of fuels above 100 octane number in the USA is measured in terms of cc (cu-cm) of TEL per US gallon of *iso*-octane. A fuel having an octane number of 110 means that that fuel has the same tendency to knock as a mixture of 10 cc of TEL in 1 US gallon of *iso*-octane. The importance of such fuels (commonly used in aero-engine) has

diminished with an introduction of turbojet turbines in aircraft.

10.26.1 Anti-knock Agents

The knock resistance tendency of a fuel can be increased by adding anti-knock agents. The anti-knock agents are substances which decrease the rate of preflame reaction by delaying the auto-ignition of the end mixture in the engine until the flame generated by spark plug passes through the end mixture.

Among all, TEL [$Pb(C_2H_5)_4$] is the most powerful antiknock agent. It has helped to improve the efficiency of the engine and to increase the specific output of SI engines. Its use will not improve the performance of the engine which is not knocking unless the spark is advanced,

the compression ratio is increased, or a higher inlet pressure is used to take the advantage of an increase in octane number. Advancing the spark will improve the performance of the engine only if optimum spark for a non-knocking fuel is not already used.

Some other anti-knock agents with their relative effectiveness are listed in Table 10.13.

Table 10.13 *Anti-knock agents*

Knocking essentially results in an incomplete and non-uniform combustion of the fuel. It is caused by the sudden and spontaneous ignition of the unburned mixture, ahead of the

flame front advancing through the combustion chamber. Partial oxidation reactions of the hydrocarbons in this mixture produce reactive intermediates that lead to such spontaneous ignition and overheating of the engine. This overheating can result in the wearing of engine parts. Antiknock agents react with these intermediates and reduce their ability to propagate the oxidation that causes knocking. Octane number of the gasoline is a measure of the effectiveness of the antiknock agent it contains. Higher this number lesser is the knock. Table 10.14 shows some of these agents and their relative effectiveness.

Table 10.14 *Relative effectiveness of antiknocks*

Gasoline composition, methods of test, and antiknock concentrations affect both the absolute and relative effectiveness of different compounds, even the same element. Moreover, some compounds can function as an antiknock under some conditions but promote knock under other. Therefore, the comparison of active elements in various compounds is representative but subject to considerable variation.

The effectiveness of an element can vary greatly, depending on the specific compound considered. Some aromatic amines are not antiknocks while others are twice as effective as N-methylamine (nitrogen in other forms is not effective).

Most metallic compounds possessing the antiknock property have metal to carbon bonds but there are exceptions. In case of organometallics, it is believed that the products released by the thermal decomposition of the compound at an appropriate stage of the precombustion process are the active antiknock species.

TEL is the most widely used (also known longest) of all antiknocks. A hydrogen atom in ethane is substituted by the heavy metal atom of lead in this organometallic compound. The average TEL usage, at present, in motor gasoline (2.0 gms lead per gallon in regular and 2.7 gms in premium) usually takes up the octane number by 6 to 10. It is a mixture of tetra methyl and ethyl leads,

and the intermediate lead alkyls produced by these two are also commercially used as antiknocks. These higher volatile alkyls offer advantages of better vapourisation and are the most effective in some multicylinder engines. Concentrated and alkyls are hazardous to handle. Therefore, these antiknock agents are blended with gasoline in refineries under same conditions; the resulting leaded gasoline can be handled as a motor fuel safely. The use of leaded gasoline, however, is not a perfect solution to the problem. It leads to the emission of lead into the atmosphere which is known to be hazardous.

10.26.2 Performance Number

The detonation tendency is also

measured in terms of performance number. The performance number (PN) is defined as

where KLIMEP represents conception of knock limited indicated mean effective pressure.

The performance number is obtained for a specified engine, under a specified set of conditions by varying the inlet pressure.

The PN method of rating enables to develop a scale beyond 100 octane number. There is a relationship between performance number and Octane number as shown in Fig. 10.93.

It has already been mentioned that by adding TEL to *iso*-octane increases knock resistance tendency. Figure 10.96 also shows the quantity of TEL by volume in millilitre per litre of gasoline on X-axis for octane numbers exceeding 100 and performance number on y-axis.

10.27 ❒ HIGHEST USEFUL COMPRESSION RATIO

In the development of SI engines, it was of great interest to use higher compression ratio as the efficiency and power output of the engine increase with an increase in compression ratio. The maximum compression ratio of any SI engine is limited by its tendency to knock. Therefore, previously, the rating of SI engine fuels was based on the highest useful compression ratio

(HUCR) that could be employed in a given engine under a given set of conditions. To find the highest compression ratio of a fuel, the compression ratio of the engine is increased (using a variable compression ratio engine) till knocking became audible with certain temperature conditions and with mixture strength and ignition, both adjusted to give highest efficiency. The HUCR of a few fuels determined by using Richardo E6 engine are listed in Table 10.15.

Figure 10.93 *Relationship between performance number and octane number*

Table 10.15 *HUCR for various fuels*

Through the measurement of HUCR of various fuels, a measure of their anti-knock value can be obtained.

The effect of anti-knock agents on HUCR is shown in Fig. 10.94.

Figure 10.94 *Effect of anti-knock agents on HUCR*

Figure 10.95 *Variation of properties of fuel with cetane number*

10.28 ❐ RATING OF CI ENGINE FUELS

Knocking is also encountered in CI engines, with an effect similar to that in SI engines, although it is due to a different phenomenon. Knock in the CI engine is due to sudden ignition and abnormally rapid combustion of accumulated fuel in the combustion chamber. Such a situation occurs because of ignition lag in the combustion of fuel between the time of injection and actual burning. As the ignition lag increases, the amount of fuel accumulated in the combustion

chamber, before starting of combustion, also increases. When combustion actually takes place, abnormal amount of energy is suddenly released, causing an excessive rate of pressure rise which results in an audible knock.

A CI engine knock can be controlled by decreasing ignition lag. The shorter an ignition lag, there is less tendency to knock. A good CI engine fuel, therefore, will ignite more readily, that is, it will have short ignition delay. Furthermore, ignition lag affects the starting, warm-up, and production of exhaust smoke in CI engines.

The property of ignition lag is generally measured in terms of *cetane number*. Cetane, a straight chain paraffin with

good ignition quality, is arbitrarily assigned a rating of 100 cetane number (CN). It is mixed with alpha-methyl naphthalene, a hydrocarbon with poor ignition quality, which is assigned zero CN. The mixture is matched with a fuel under test in standard (CFR) engine running under prescribed conditions. The CN of the fuel is then defined as the percent by volume of cetane in a mixture of cetane and alpha-methyl-naphthalene that produces the same ignition lag as the fuel being tested, in the same engine and under the same operating conditions.

If a test fuel has a CN of 40, it means a mixture containing 40% cetane and 60% a-methyl naphthalene by volume gives

the same ignition delay as tested fuel.

The straight chain paraffins are the most suitable fuels for CI engines.

The graph shown in Fig. 10.95 shows the relationship of the other properties of the fuel with CN which are also responsible for the knocking of the engine.

10.29 ❒ IC ENGINE FUELS

The desirable properties of IC engine fuels are as follows:

1. High energy density
2. Good combustion qualities
3. High thermal stability
4. Low deposit forming tendencies
5. Compatibility with the engine hardware
6. Good fire safety
7. Low toxicity
8. Low pollution
9. Easy transferability on board vehicle storage

10.29.1 Fuels for SI Engines

Gasoline

It is a mixture of various hydrocarbons such as paraffins, olefins, napthalenes, and aromatics. The requirements of an ideal gasoline are as follows:

1. It should mix readily with air and afford uniform manifold distribution, *i.e.*, it should easily vapourise.
2. It must be knock-resistant.
3. It should not pre-ignite easily.
4. It should not tend to decrease the volumetric efficiency of engine
5. It should be easy to handle.
6. It should be cheap and easily available.
7. It should not corrode engine parts.
8. Its calorific value should be high.
9. It should not form gum and varnish.

Knock Rating of SI Engine Fuels

Highest Useful Compression Ratio: The HUCR is the highest compression ratio at which a fuel can be used without detonation in a specified test engine under specified operating conditions and

the ignition and mixture strength being adjusted to give best efficiency. The HUCR of different fuels is given in Table 10.16.

Rating of SI Engine Fuels

The rating of fuel is done by comparing its performance with that of a standard reference fuel which is usually a combination of *iso*-octane and *n*-heptane. *Iso*-octane offers great resistance to deformations and is arbitrarily assigned a rating of 100-octane number. On the other hand, n-heptane detonates very rapidly, and is assigned a rating of zero-octane number.

Octane Number

It is percentage of *iso*-octane by volume

in a mixture is *iso*-octane and n-heptane, which exactly matches the knocking intensity of a given fuel, in a standard engine under given standard operating conditions.

The octane scale is extended beyond 100 by adding TEL to *iso*-octane.

Table 10.16 *HUCR for different fuels*

10.29.2 Fuels for CI Engines

Diesel fuels are used for CI engines. These are petroleum fractions that lie between kerosene and lubricating oils.

The main desirable characteristics of diesel fuels are as follows:

1. Cleanliness—carbon residue, contamination, sulphur, etc.
2. Ignition quality—cetane number
3. Fluidity—viscosity, pour point, etc.
4. Volatility—flash point and carbon residue

Rating of CI Engine Fuels

The rating of a diesel fuel is measured in terms of its cetane number. Cetane, a straight chain paraffin with good ignition quality, is arbitrarily assigned a rating of 100-cetane number. It is mixed with a-methyl naphthalene, a hydrocarbon with poor ignition quality, which is assigned zero-cetane number.

Cetane Number

This is defined as the percentage of volume of cetane in a mixture of cetane and α-methyl naphthalene that produces the same ignition lag as the fuel being tested, in the same engine and under the same operating conditions.

The various alternative fuels for IC engines are as follows:

1. Alcohols: ethanol and methanol
2. Hydrogen
3. Biogas
4. Producer (or water) gas
5. Biomass generated gas
6. LPG and LNG
7. CNG
8. Coal gasification and coal liquefication
9. Non-edible vegetable oils
10. Non-edible wild oils
11. Ammonia

10.30.1 Alcohols

Alcohols are of two types—ethanol (C_2H_5OH) and methanol (CH_3OH) which can be produced from sugarcane waste and other agricultural products. Ethanol can be produced by the termination of vegetables and plant materials such as sugarcane, molasses, starchy matter, cellulose material (wood, sulphite, waste liquor from paper

manufacture), hydrocarbon gases, and so on.

The use of ethanol in IC engines gives better performance than on a gasoline engine. The power output, BSHC, maximum thermal efficiency, and engine torque are higher as compared to petrol engines. Alcohols are not suitable fuels for CI engines.

Methanol behaves much like petroleum. It can be used directly or mixed with gasoline. Some of the important features of methanol as fuel are as follows:

1. The specific fuel consumption with methanol as fuel is 50% less than petrol engine.
2. Exhaust CO and HC are decreased continuously with blends.
3. Methanol can be used as supplementary fuel in heavy vehicles.

Methanol can be produced from coal,

natural gas, oil shell, farm waste, etc. Methanol emits less amount of CO_2 and other polluting gases as compared to gasoline-fuelled vehicles. However, methanol has high latent heat and gives cold starting problems, is more corrosive, and more toxic as compared to petrol, gives high levels of formaldehyde emission. The important objectionable emissions are CO, HC, NO_x, and aldehydes.

10.30.2 Use of Hydrogen in CI Engines

There are two methods by which hydrogen can be used in CI engines as follows.

1. By introducing hydrogen with air and using a spray of diesel oil to ignite the mixture by the dual fuel mode. The limiting conditions are when the diesel quantity is too small to produce effective ignition, that is failure of ignition and when the hydrogen air mixture is so rich that the combustion becomes violent. In between these limits, a wide range of diesel-to-

hydrogen proportions can be tolerated.

2. By introducing hydrogen directly into the cylinder at the end of compression. Since the self-ignition temperature of hydrogen is very high, the gas spray is made to impinge on a hot glow plug in the combustion chamber by surface ignition. It is also possible to feed a very lean hydrogen air mixture during the intake into an engine and then inject the bulk of the hydrogen towards the end of the compression stroke.

Advantages of Hydrogen as IC Engine Fuel

1. **Low emission:** Essentially, no CO or HC in the exhaust as there is no carbon in the fuel. Most exhaust would be H_2O and N_2.
2. **Fuel availability:** There are a number of different ways of making hydrogen, including electrolysis of water.
3. Fuel leakage to environment is not a pollutant.
4. High energy content per volume when stored as a liquid. This would give a large range vehicle range for a given fuel tank capacity.

Disadvantages of Hydrogen as IC Engine Fuel

1. Requirement of heavy, bulky fuel storage both in vehicle and at the service station. Hydrogen can be stored as a cryogenic liquid or as a compressed gas. If stored as a liquid, it would have to be kept under pressure at a very low temperature. This would require a thermally super-insulated fuel tank. Storing in a gas phase would require a high pressure vessel with limited capacity.
2. Difficult to refuel.
3. Poor engine volumetric efficiency. When a gaseous fuel is used in an engine, the fuel will displace some of the inlet air and less volumetric efficiency will result.
4. Fuel cost would be high in the context of present–day technology and availability.
5. High NO_x emissions because of high flame temperature.
6. Can detonate.

10.30.3 Biogas

Biogas is produced from organic waste (cow dung) which is easily available. The primary advantages of biogas is its ability to operate the engine on lean mixture, which reduces exhaust HC and CO_2 concentration in the engine exhaust. It gives less deposits and shows clean burning characteristics as compared to petrol and diesel oil.

SI engines can be operated on biogas after starting the engine by using petrol. Biogas can be used in SI engine in two forms as follows:

1. To run the engine entirely on biogas
2. Dual fuel engine operation on biogas and petrol

The advantages of using biogas as fuel in a CI engine are as follows:

1. The gas-air mixture provides uniform mixture in a multi-cylinder engine at all times.
2. There is virtually no CO emission in exhaust due to lean operation of the engine.
3. NO_x emissions are reduced by about 60%.
4. Soot is virtually eliminated and exhaust is found to have less pungent odour than diesel oil.

10.30.4 Producer (or Water) Gas

Producer gas is a mixture of CO, H_2, and N_2. It is obtained from the partial oxidation of coal, or peat when burnt with insufficient quantity of air. In using the producer gas in an SI engine, the problem remains as to how and with which fuel it has to be produced economically. The producer gas can be used with an SI engine but with a lesser output.

10.30.5 Biomass-generated Gas

Biomass comprises all kinds of agricultural waste such as stalks, husks

of different crops, residues like bagasse, and so on. Biomass gasification gives producer gas.

10.30.6 LPG as SI Engine Fuel

LPG consists of hydrocarbons of such volatility that they exist as gases under atmospheric conditions, but can be readily liquefied under pressure. LPG contains mainly members of paraffin hydrocarbons up to C_4. It consists of propane and butane, to some extent.

Advantages of LPG in SI Engine

1. LPG contains less carbon than petrol. Therefore, emission is much reduced by its use.
2. LPG mixes with air at all temperatures.
3. In multi-cylinder engines, a uniform mixture can be supplied to all cylinders.
4. There is no crankcase dilution because the fuel is in the form of vapour.
5. Automobile engines can use propane if they have high compression ratio.
6. LPG has high antiknock characteristics.
7. Its heat energy is about 80% of gasoline, but its high octane value compensates the thermal efficiency of the engine.

8. Running on LPG translates into a cost saving of about 50%.
9. The engine may have a 50% longer life.

Disadvantages of LPG in SI Engine

1. Engines are normally designed to take in a fixed volume of the mixture and air. Therefore, LPG will produce 10% less power for a given engine at full throttle.
2. The ignition temperature of LPG is somewhat higher than petrol. Therefore, running on LPG could lead to a 5% reduction in valve life.
3. A good cooling system is quite necessary, because LPG vapouriser uses engine coolant to provide the heat to convert the liquid LPG to gas.
4. The vehicle weight is increased due to the use of heavy pressure cylinders for storing LPG.
5. A special fuel feed system is required for LPG.

Liquefied natural gas (LNG) comes from dry natural reservoirs, mainly CH_4 (methane) with a very small percentage of ethane and propane. The major difficulty in using this gas is its very low boiling point (–161.5°C)

10.30.7 Compressed Natural Gas

Compressed natural gas (CNG) is composed of methane, ethane, and

propane with traces of other gases. Methane is the main constituent. It has very good anti-knock qualities and burns at much higher temperature as compared with petrol. It can be safely used in engines with a compression ratio up to 12:1. It gives higher thermal efficiency as compared to gasoline engine. The pollutants emitted by CNG are also reduced. It is nontoxic and has been widely used in three-wheelers and buses.

Advantages of CNG

1. It is cheaper than petrol.
2. It is engine friendly–gives lower maintenance cost and longer life.
3. It is safe–it is lighter than air and safely dissipates to atmosphere.
4. It mixes easily with air completely and gives proper combustion.
5. It is odourless.

10.30.8 Coal Gasification and Coal Liquefaction

These fuels are mainly used for power generation than IC engines.

10.30.9 Non-edible Vegetable Oils

The vegetable oils including soya bean oil, cotton seed oil, sunflower oil, rapeseed oil, palm oil, coconut oil, and linseed oil have cetane numbers and calorific values comparable with those of diesel oil and can be used as alternative fuels for diesel engine. The main disadvantage is their high viscosity which gives difficulties with fuel injection and cold flow pumping.

10.30.10 Non-edible Wild Oils

Of all the non-edible wild oils, mahua and neem oils have the highest production. Their use as IC engine fuels

has not become popular yet.

10.30.11 Ammonia

Ammonia has been demonstrated as a practical fuel for engine applications. Ammonia does not contain carbon and hence, there is an absence of two major pollutants, HC and CO, in the exhaust. These are its added advantages. The performance characteristics of engine using NH_3 as fuel are similar to the performance characteristics obtained with petroleum fuels.

The pros and cons of alternative fuels are given in Table 10.17.

Table 10.17 *Pros and cons of alternative fuels*

Summary for Quick Revision

1. An internal combustion engine is a device in which the fuel is mixed with air and burned inside the engine to generate power.
2. The IC engines can be either of spark ignition or compression ignition type.
3. The scavenging process refers to clearing the cylinder after the expansion stroke with fresh-air charge.
4.
5. The process of mixing air and fuel, outside the engine cylinder, in proper proportions, is called carburetion.
6. For a simple carburettor:
    1. Approximate analysis:
        1. Mass of air per second,
        2. Mass of fuel per second,

           A_f = area of fuel nozzle, C_d = coefficient of discharge.

        3. For $z = 0$,
    2. Exact analysis:
        1.
        2.
        3.
7. There are two mechanically operated petrol injection systems —combustion chamber injection and continuous port injection. The first method is not being used nowadays.
8. Fuel injection systems for CI engine are air injection system and solid injection.
9. Fuel nozzle design:
    1. Fuel consumed per cycle,
    2. Duration of injection in seconds,
    3. Mass of fuel supplied per second,
    4. Volume of fuel injected per second,

   where d = orifice diameter, n = number of

orifices, θ = crank angle duration, N_i = number of injections per minute, $N = rpm$, C_{df} = coefficient of discharge of fuel nozzle.

10. Fuel ignition systems are battery ignition system, magneto ignition system, and electronic ignition system.
11. Ignition lag (or delay) is the phase during which some fuel has already been admitted but has not been ignited.
12. The phenomenon of detonation occurs in SI engines when the temperature of unburned mixture exceeds the self-ignition temperature of the fuel during the ignition lag. This gives rise to spontaneous or auto-ignition of fuel at various pin-point locations.
13. The phenomenon of pulsating variation of pressure due to sudden burning of accumulated fuel during the delay period in CI engines is called knocking.
14. There are two types of lubrication systems—splash lubrication systems and pressure feed lubrication system.
15. There are mainly two types of cooling systems—air cooling system and water cooling system.
16. There are four methods used for governing of IC engines—hit and miss governing, qualitative governing, quantitative governing, and combination methods.

Multiple-choice Questions

1. An Otto cycle on internal energy (U) and entropy (s) diagram is shown in:
    1.
    2.
    3.
    4.
2. Which of the following statements is “true”?
    1. The term “Knock” is used for on identical phenomenon in a spark ignition and compression ignition engine.
    2. “Knock” is a term associated with a phenomenon taking place in the early part of combustion in a spark ignition engine and the later part of combustion in a spark ignition engine.

    3. “Knock” is a term associated with a phenomenon taking place in the early part of combustion in a spark ignition engine and the later part of combustion in a compression ignition engine.
    4. None of the above
3. The essential function of the carburettor in a spark ignition engine is to
    1. meter the fuel into air stream and amount dictated by the load and speed
    2. bring about mixing of air and fuel to get a homogeneous mixture
    3. vapourise the fuel
    4. distribute fuel uniformly to all cylinders in a multi-cylinder engine and also vapourise it.
4. Match List I with List II and select the correct answer using the codes given below the lists:

Codes:

A B C D

    1. 4 5 3 2
    2. 1 3 5 2
    3. 2 3 1 5
    4. 4 1 2 3
5. Which of the following factors increase detonation in the SI engine?
    1. IncrEase spark advance
    2. Increased speed
    3. Increased air-fuel ratio beyond stoichiometric strength
    4. Increased compression ratio

Select the correct answer using the codes given below:

Codes:

    1. I and III
    2. II and IV
    3. I, II, and IV
    4. I and IV
6. Which one of the following curves is proper representation of pressure differential (*y*-axis) vs velocity of air (*x*-axis) at the throat of a carburettor?
    1.
    2.
    3.
    4.
7. Match List I with List II and select the correct answer using the codes given below the lists:

    Codes:

    A B C D

    1. 1 2 3 4
    2. 1 3 4 2
    3. 2 3 4 1
    4. 4 1 2 3
8. Match List I with List II and select the correct answer using the codes given below the lists:

    Codes:

    A B C

    1. 1 2 3
    2. 1 3 2
    3. 2 3 1
    4. 3 1 2
9. The air-fuel ratio for idling speed of automobile petrol engine is closer to

1. 10:1
2. 15:1
3. 17:1
4. 21:1

10. Consider the following statements:
    1. The performance of an SI engine can be improved by increasing the compression ratio.
    2. Fuels of higher octane number can be employed at higher compression ratio.

    Of these statements

    1. both I and II are true
    2. both I and II are false
    3. I is true but II is false
    4. I is false but II is true
11. The object of providing masked inlet valve in the air passage of compression-ignition engines is to
    1. enhance flow rate
    2. control in flow
    3. induce primary swirl
    4. induce secondary turbulence
12. Which one of the following events would reduce the volumetric efficiency of a vertical compression ignition engine?
    1. Inlet valve closing after bottom head centre
    2. Inlet valve closing before bottom dead centre
    3. Inlet valve opening before top dead centre
    4. Exhaust valve closing after top dead centre
13. As compared to air standard cycle, in actual working, the effect of variation in specific heats is to
    1. increase maximum pressure and maximum temperature
    2. reduce maximum pressure and maximum temperature
    3. increase maximum pressure and decrease maximum temperature
    4. decrease maximum pressure and increase maximum temperature
14. Reference fuels for knock rating of SI engine fuels would include
    1. iso-octane and alpha-methyl naphthalene
    2. normal octane and aniline
    3. iso-octane and n-hexane

4. n-heptane and so-octane

15. Consider the following measures:
    1. Increasing the compression ratio
    2. Increasing the intake air temperature
    3. Increasing the length of dimension of the cylinder
    4. Increasing the engine speed

    The measures necessary to reduce the tendency to knock in CI engines would include

    1. I, II, and III
    2. I, II, and IV
    3. I, III, and IV
    4. II, III, and IV
16. Match List I with List II and select the correct answer using the codes given below the lists:

    Codes:

    A B C D

    1. 2 1 3 4
    2. 1 2 4 3
    3. 2 1 4 3
    4. 1 2 3 4
17. Generally, in Bosch type fuel injection pumps, the quantity of fuel is increased or decreased with change in load, due to change in
    1. timing of start of fuel injection
    2. timing of end fuel injection
    3. injection pressure of fuel
    4. velocity of flow of fuel
18. Match List I with List II and select the correct answer using the codes given below the lists:

Codes:

	A	B	C	D
1.	2	4	5	1
2.	1	3	4	2
3.	5	1	1	2
4.	2	5	3	1

19. Knocking in the SI engine decreases in which one of the following orders of combustion chamber designs?
    1. F head, L head, I head
    2. T head, L head, F head
    3. I head, T head, F head
    4. F head, I head, T head
20. The two reference fuels used for cetane rating are
    1. cetane and isooctane
    2. cetane and tetraethyl lead
    3. cetane and n-heptane
    4. cetane and a-methyl naphthalene
21. Match List I with II, in respect of SI engines, and select the correct answer using the codes given below the lists:

Codes:

	A	B	C	D
1.	2	3	1	4
2.	3	2	1	4
3.	2	3	4	1
4.	3	4	2	1

22. By higher octane number of SI fuel, it is mean that the fuel has
    1. higher heating value
    2. higher flash point
    3. lower volatility
    4. longer ignition delay
23. Which of the following factors would increase the probability of knock in the CI engines?
    1. Long ignition delay of fuel

2. Low self-ignition temperature of fuel
3. Low volatility of fuel

Select the correct answer using the codes given below:

Codes:

1. I, II, and III
2. I and II
3. I and III
4. II and III

24. List I gives the different terms related to combustion while List II gives the outcome of the events that follow. Match List I with List II and select the correct answer.

Codes:

A B C D

1. 3 4 1 2
2. 4 3 1 2
3. 3 4 2 1
4. 4 3 2 1

25. Which one of the following engines will have heavier flywheel than the remaining ones?
    1. 40 kW four-stroke petrol engine running at 1500 rpm
    2. 40 kW two-stroke petrol engine running at 1500 rpm
    3. 40 kW two-stroke diesel engine running at 750 rpm
    4. 40 kW four-stroke diesel engine running at 750 rpm
26. Consider the following statements:

Knock in the SI engine can be reduced by

1. supercharging
2. retarding the spark
3. using a fuel of long straight chain structure

4. increasing the engine speed

Of these statement

1. I and II are correct
2. II and III are correct
3. I, III, and IV are correct
4. II and IV are correct

27. Consider the following statements:

The injector nozzle of a CI engine required to inject fuel at a sufficiently high pressure in order to

1. be able to inject fuel in a chamber of high pressure at the end of the compression stroke.
2. inject fuel at high velocity to facilitate atomisation.
3. ensure that penetration is not high.

Of these statements

1. I and II are correct
2. I and III are correct
3. II and III are correct
4. II and IV are correct

28. Match List I with List II select the correct answer:

Codes:

A B C D

1. 4 3 2 1
2. 2 4 1 3
3. 4 2 1 3
4. 2 4 3 1

29. Compensating jet in carburettor supplies almost constant amount of petrol at all speeds because

1. the jet area is automatically varied depending on the suction.
2. the flow from the main jet is diverted to the compensating jet with increase in speed.
3. the diameter of the jet is constant and the discharge coefficient is invariant.
4. the flow is produced due to the static head in the float Chamber.

30. If methane undergoes with the stoichiometric quantity of air, then the air-fuel ratio on molar basis would be
    1. 15.22:1
    2. 12.30:1
    3. 14.56:1
    4. 9.52:1
31. For maximum specific output of a constant volume cycle (Otto cycle)
    1. the working fluid should be air
    2. the speed should be high
    3. suction temperature should be high
    4. temperature of the working fluid at the end of compression and expansion should be equal
32. In a SI engine, which one of the following is the correct order of the fuels with increasing detonation tendency?
    1. Paraffins, olefins, naphthenes, and aromatics
    2. Aromatics, naphthenes, paraffins, and olefins
    3. Naphthenes, olefins, aromatics, and paraffins
    4. Aromatics, naphthenes, olefins, and paraffins
33. Consider the following statements:

    Detonation in the SI engine can be suppressed by

    1. retarding the spark timing
    2. increasing the engine speed
    3. using 10% rich mixture

    Of these statements

    1. I and III are correct
    2. II and III are correct
    3. I, II, and III are correct
    4. I and II are correct
34. For the same maximum pressure and temperature

1. Otto cycle is more efficient than diesel cycle
2. diesel cycle is more efficient than Otto cycle
3. dual cycle is more efficient than Otto and diesel cycles
4. dual cycle is less efficient than Otto and diesel cycles

35. Velocity of flame propagation in the SI engine is maximum for a fuel-air mixture which is
    1. 10% richer than stoichiometric
    2. equal to stoichiometric
    3. more than 10% richer than stoichiometric
    4. 10% leaner than stoichiometric
36. Divided chamber diesel engines use lower injection pressure compared to open chamber engines because
    1. pintle nozzles cannot withstand high injection pressures
    2. high air swirl does not require high injection pressures for atomisation
    3. high injection pressures may cause overpenetration
    4. high injection pressure causes leakage of the fuel at the pintle
37. For the same maximum pressure and heat input, the most efficient cycle is
    1. (a) Otto cycle
    2. (b) Diesel cycle
    3. (c) Brayton cycle
    4. (d) Dual combustion cycle
38. The order of values of thermal efficiency of Otto, diesel, and dual cycles, when they have equal compression ratio and heat rejection, is given by
    1. $\eta_{otto} > \eta_{diesel} > \eta_{dual}$
    2. $\eta_{diesel} > \eta_{dual} > \eta_{otto}$
    3. $\eta_{dual} > \eta_{diesel} > \eta_{otto}$
    4. $\eta_{otto} > \eta_{dual} > \eta_{diesel}$
39. In an air-standard Diesel cycle, r is the compression ratio, ρ is the fuel cut-off ratio and g is the adiabatic γ, its air standard efficiency is given by
    1.
    2.
    3.
    4.
40. Consider the following statements:
    1. Octane rating of gasoline is based on isooctane and iso-heptane fuels which are paraffins.

    2. Tetraethyl lead is added to gasoline to increase octane number.
    3. Ethylene dibromide is added as scavenging agent to remove lead deposits on spark plugs.
    4. Surface ignition need not necessarily cause knocking.

    Which of these statements are corrects?

    1. I, II, III, and IV
    2. II, III, and IV
    3. I and IV
    4. I, II, and III

41. Consider the following statements:
    1. Recycling exhaust gases with intake increases emission of oxides of nitrogen from the engine.
    2. When the carburettor throttle is suddenly opened, the fuel air mixture leans our temporarily causing engine stall.
    3. The effect of increase in altitude on carburettor is to enrich the entire part throttle operation.
    4. Use of multiple venturi system makes it possible to obtain a high velocity air stream when the fuel is introduced main venturi throat.

    Which of these statements are correct?

    1. I and III
    2. I and II
    3. II and III
    4. II and IV

42. Match List I (Air-fuel ratio by mass) with List II (Engine operation mode) and select the correct answer using the codes given below the lists:

    Codes:

    A B C D

1. 3 2 1 5
2. 4 2 1 5
3. 3 1 2 4
4. 4 1 2 3

43. Consider the following statements:

In down draft carburettor, a hot spot is formed at the bottom wall which is common for intake and exhaust manifolds. This helps to

1. improve evaporation of liquid fuel
2. provide higher thermal efficiency
3. reduce fuel consumption
4. lower the exhaust gas temperature

Which of these statements are correct?

1. I, II, and IV
2. I, II, and III
3. I, III, and IV
4. II, III, and IV

44. In a petrol engine car, which one of the following performance characteristics is affected by the front-end volatility of the gasoline used?
    1. Hot starting and vapour lock
    2. Engine warm-up and spark plug fouling
    3. Spark plug fouling and hot starting
    4. Vapour lock, engine warm-up, and spark plug fouling
45. In the operation of four-stroke diesel engines, the term ‘squish’ refers to the
    1. injection of fuel in the pre-combustion chamber
    2. discharge of gases from the pre-combustion chamber
    3. entry of air into the combustion chamber
    4. stripping of fuel from the core
46. Consider the following statements regarding the advantages of fuel injection over carburetion in SI engines:

1. Higher power output and increased volumetric efficiency.
2. Simple and inexpensive injection equipment.
3. Longer Life of injection equipment.

4. Less knocking and reduced tendency for back-fire.

Select the correct answer using the codes given below:

1. I, II, and III
2. I, II, and III
3. II and III
4. I and IV

47. Stoichiometric air-fuel ratio by volume for combustion of methane in air is
    1. 15:1
    2. 17.16:1
    3. 9.52:1
    4. 10.58:1
48. Auto-ignition time for petrol-air mixture is minimum when the ratio of actual fuel-air ratio and chemically correct fuel-air ratio is
    1. 0.8
    2. 1.0
    3. 1.2
    4. 1.5
49. Consider the following statements regarding knock rating of SI engine fuels:
    1. Iso-octane is assigned a rating of zero octane number.
    2. Normal heptane is assigned a rating of hundred octane number.
    3. Iso-octane is assigned a rating of hundred octane number.
    4. Normal heptane is assigned a rating of zero octane number.

Which of the above statements are correct?

1. I and II
2. II and III
3. III and IV
4. IV and I

50. In spark ignition engines knocking can be reduced by
    1. increasing the compression ratio

2. increasing the cooling water temperature
3. retarding the spark advance
4. increasing the inlet air temperature

51. The tendency of knocking in CI engine reduces by
    1. high self-ignition temperature of fuel
    2. decrease in jacket water temperature
    3. injection of fuel just before TDC
    4. decrease in injection pressure
52. Consider the following statements relevant to the ignition system of SI engine:
    1. Too small a dwell angle will lead to the burning of condenser and contact points.
    2. Too small a dwell angle will result in misfiring.
    3. Too large a dwell angle will result in burning of condenser and contact points.
    4. Too large a dwell angle will result in misfiring.

    Which of the above statements are correct?

    1. I and II
    2. II and III
    3. III and IV
    4. IV and I
53. Consider the following statements in respect of automobile engine with thermosyphon cooling:
    1. Heat transfer from gases to cylinder walls takes place by convection and radiation.
    2. Most of the heat transfer from radiator to atmosphere takes place by radiation.
    3. Most amount of heat transfer from radiator to atmosphere takes place by convection.
    4. Heat transfer from cylinder walls take place by conduction and convection.

    Which of the above statements are correct?

    1. I, II and IV
    2. I, III, and IV
    3. II, III, and IV
    4. I and II
54. Which of the following action(s) increase(s) the knocking tendency in the SI engine?
    1. Increasing mixture strength beyond equivalence ratio

ratio (ϕ) = 1.4

2. Retarding the spark and increasing the compression ratio.
3. Increasing the compression ratio and reducing engine speed.
4. Increasing both mixture strength beyond equivalence ratio (ϕ) = 1.4 and the compression ratio

55. Which of the following features(s) is/are used in the combustion chamber design to reduce SI engine knock?
    1. Spark plug located away from exhaust valve, wedge-shaped combustion chamber, and short flame travel distance
    2. Wedge-shaped combustion chamber
    3. Wedge-shaped combustion chamber and short flame travel distance
    4. Spark plug located away from exhaust valve, short flame travel distance, and side valve design
56. Which of the following factor(s) increase(s) the tendency for knocking in the CI engine?
    1. Increasing both the compression ratio and the coolant temperature
    2. Increasing both the speed and the injection advance
    3. Increasing the speed, injection advance and coolant temperature
    4. Increasing the compression ratio
57. Match List I with List II and select the correct answer using the codes given below the lists:

Codes:

	A	B	C	D
1.	2	4	3	1
2.	5	4	1	3
3.	2	3	5	1
4.	5	3	1	4

58. Consider the following statements for a multi-jet carburettor:
    1. Acceleration jet is located just behind the throttle valve.

2. Idle jet is located close to the choke.
3. Main jet alone supplies petrol at normal engine speeds.

Which of the statements given above are correct?

1. I, II, and III
2. I and II
3. II and III
4. I and III

59. The stoichiometric air-fuel ratio for petrol is 15:1. What is the air-fuel ratio required for maximum power?
    1. 16:1 – 18:1
    2. 15:1
    3. 12:1 – 14:1
    4. 9:1 – 11:1
60. For which of the following reasons, do the indirect injection diesel engines have higher specific output compared to direct injection diesel engines?
    1. They have lower surface to volume ratio.
    2. They run at higher speeds.
    3. They have higher air utilization factor.
    4. They have lower relative heat loss.

Select the correct answer using the code given below:

1. I and II
2. II only
3. II and III
4. III and IV

61. Consider the following statements:
    1. In a carburettor, the throttle valve is used to control the fuel supply.
    2. The fuel level in the float chambers is to be about 4 to 5 mm below the orifice level of main jet.
    3. An idle jet provides extra fuel during sudden acceleration.
    4. A choke valve restricts the air supply to make the gas richer with fuel.

Which of the statements given above are correct?

1. II and IV
2. I and III
3. I, II, and III
4. II, III, and IV

62. The knocking tendency in compression ignition engines increases with
    1. increase of coolant water temperature
    2. increase of temperature of inlet air
    3. decrease of compression ratio
    4. increase of compression ratio
63. Which of the following cannot be caused by a hot spark plug?
    1. Pre-ignition
    2. Post-ignition
    3. Detonation
    4. Run-on-ignition

    Select the correct answer using the code given below:

    1. I and IV
    2. II only
    3. II and III
    4. III only
64. Which of the following combustion chamber design features reduce(s) knocking in SI engines?
    1. Spark plug located near the inlet valve
    2. T-Head
    3. Wedge-shaped combustion chamber
    4. Short flame travel distance

    Select the correct answer using the code given below:

    1. I and III
    2. III only
    3. III and IV
    4. I and II
65. A 4-stroke diesel engine, when running at 2000 rpm has an injection duration of 1.5 ms. What is the corresponding duration of the crank angle in degrees?
    1. 18°
    2. 9°

3. 36°
4. 15°

66. Consider the following statements:
    1. In the SI engines, detonation occurs near the end of combustion, whereas in CI engines, knocking occurs near the beginning of combustion.
    2. In SI engines, no problems are encountered on account of preignition.
    3. Low inlet pressure and temperature reduce knocking tendency in SI engines but increase the knocking tendency in CI engines.

    Which of the statements given above are correct?

    1. I, II, and III
    2. Only I and II
    3. Only II and III
    4. Only I and III
67. The tendency of petrol to detonate in terms of octane number is determined by comparison of fuel with which of the following?
    1. Iso-octane
    2. Mixture of normal heptane and iso-octane
    3. Alpha methyl naphthalene
    4. Mixture of methane and ethane
68. For the same indicated work per cycle, mean speed and permissible fluctuation of speed, what is the size of flywheel required for a multi-cylinder engine in comparison to a single-cylinder engine?
    1. Bigger
    2. Smaller
    3. Same
    4. Depends on thermal efficiency of the engine
69. Consider the following statements:
    1. For a diesel cycle, the thermal efficiency decreases as the cut-off ratio increases.
    2. In a petrol engine, the high voltage for spark is in the order of 1000 V.
    3. The material for centre electrode in spark plug is carbon.

    Which of the statements given above is/are correct?

1. Only I
2. Only I and II
3. Only II and III
4. I, II, and III

70. Consider the following statements:

    In order to prevent detonation in a spark ignition engine, the charge away from the spark plug should have:

    1. low temperature
    2. low density
    3. long ignition delay

    Which of the statements given above is/are correct?

    1. Only I
    2. Only II
    3. Only III
    4. I, II, and III
71. Where does mixing of fuel and air take place in case of diesel engine?
    1. Injection pump
    2. Injector
    3. Engine cylinder
    4. Inlet manifold
72. Consider the following statements regarding cetane:
    1. It is standard fuel used for knock rating of diesel engines.
    2. Its chemical name is n-hexadecane.
    3. It is a saturated hydrocarbon of paraffin series.
    4. It has long carbon chain structure.

    Of the above statements

    1. I, III, and IV are correct
    2. I, II, and III are correct
    3. I, II, and IV are correct
    4. II, III, and IV are correct

Review Questions

1. Define an engine.
2. Define a heat engine.
3. How IC engines are classified?
4. Define an internal combustion engine.
5. What is an external combustion engine?
6. List the applications of IC engines.
7. List four advantages of internal combustion engine over external combustion engine.
8. Describe the salient features of SI engine.
9. Differentiate between SI and CI engines.
10. What are the main differences between 4-stroke and 2-stroke engines?
11. Explain the working of:
    1. Four-stroke SI engine
    2. Four-stroke CI engine
12. Explain the working of 2-stroke SI engine.
13. Compare 4-stroke and 2-stroke engines.
14. Compare SI and CI engines.
15. List the merits of two-stroke engines over 4-stroke engines.
16. Draw the *p-v* diagram for (a) 4-stroke petrol engine (b) 2-stroke petrol engine.
17. Why the actual *p-v* diagram for a 4-stroke diesel engine differ from the theoretical diagram?
18. Explain the working of 2-stroke diesel engine.
19. What is the function of flywheel in an internal combustion engine?
20. What is the necessity of valve-timing diagram?
21. What is scavenging process in IC engines?
22. List four applications of Wankel engine.
23. Why do actual *p-v* diagrams differ from the theoretical diagrams of IC engines?
24. Define diagram factor.
25. Write four assumptions generally made while drawing the theoretical *p-v* diagram for 2-stroke petrol engine.
26. Suggest the IC engines used for the following:
    1. Bus
    2. Motor cycle
    3. Scooter
    4. Moped
27. What are the essential requirements of an ideal carburettor?
28. What do you understand by compensation? Explain.

29. Explain with a neat diagram how the power and efficiency of an SI engine vary with A:F ratio.
30. Why are rich mixtures required for starting and during idling of an engine? Explain with the help of a neat sketch.
31. Draw a simple type of carburettor and explain its working.
32. What do you understand by stoichiometric A:F ratio?
33. What are the requirements of a good injection system?
34. What is solid injection? What are its advantages over air-injection system?
35. What are the basic requirements of an ideal ignition system?
36. Explain the working of a battery ignition system with the help of a neat sketch.
37. What are the main drawbacks of a battery ignition system? How can these be overcome?
38. Explain the working of a magneto ignition system with the help of a neat sketch.
39. Compare battery and magneto ignition systems.
40. Describe the working of an electronic ignition system.
41. What are the lubricating systems for IC engines?
42. Differentiate between splash and pressure lubrication system and describe their operations with the help of neat diagrams.
43. What is meant by dry and wet sump lubrication? Where is dry sump preferred? Why?
44. Explain the lubrication of the following parts:
    1. Crank pin and gudgeon pin
    2. Main bearing
    3. Cylinder and piston
    4. Exhaust valve
45. What is the necessity of cooling system for an IC engine?
46. Name the cooling systems generally used for I.C. engines.
47. What is the necessity of I.C. engine cooling?
48. Discuss the types of cooling systems for an IC engine.
49. What are the effects of under cooling and over cooling of an engine?
50. Draw a neat diagram of thermostatic-controlled water cooling system and explain its working.
51. Discuss the merits and demerits of water cooling with air cooling system.
52. Define the function of a radiator. Discuss different types of matrices used with these radiators.
53. What is the function of fan in radiator system? Discuss its performance with change of speed and load on the engine.
54. What do you understand by evaporative cooling? When is it

preferred?

55. What is the difference between open system and close system cooling used for IC engines? Discuss their relative merits and demerits.
56. Discuss the relative merits and demerits of air cooling system over water cooling system.
57. What is overcooling? What are its disadvantages?
58. Define combustion phenomenon in I.C. engines.
59. What are the stages of combustion in S.I. engines?
60. What is ignition lag?
61. List four factors affecting delay period.
62. List four factors which affect the velocity of flame propagation.
63. What is the phenomenon of knocking in S.I. engines?
64. List four factors which reduce knocking in S.I. engines.
65. What are the effects of knocking?
66. Write four design principles for combustion chambers.
67. What are the stages of combustion in C.I. engines?
68. Write four factors which reduce delay period in C.I. engines.
69. What is diesel knock?
70. What is knocking in C.I. engines?
71. List four factors which affect knocking in C.I. engines.
72. Give the names of few anti-knock agents.

Exercises

10.1 The venturi of a single jet carburettor has throat diameter of 75 mm and fuel nozzle diameter of 5 mm. Find the A:F ratio of the mixture supplied to the engine for the following data: $C_{da} = 0.93$, $C_{df} = 0.68$, Δp across venturi = 0.15 bar, $\rho_a = 1.29$ kg/m^3, $\rho_f =$

720 kg/m^3. Neglect nozzle lip and air compressibility effect.

[**Ans.** 13.3:1]

10.2 A four-cylinder, four-stroke petrol engine of 82.5 mm diameter and 115 mm stroke has 80% volumetric efficiency, when running at 3000 rpm. The pressure head causing the flow is 11.03 cm of Hg. Determine (a) the size of the venturi, and (b) the fuel nozzle diameter if the A:F ratio of mixture supplied is 14:1. Take $C_{da} = 0.84$, $C_{df} = 0.7$, $\rho_a = 1.29$ kg/m^3, $\rho_f = 700$ kg/m^3.

[**Ans.** 22.3 mm, 4.15 mm]

10.3 The diameter of a simple venturi is 75 mm and fuel nozzle diameter is 5 mm. Nozzle lip = 5 mm, and $C_{da} = 0.85$, $C_{df} = 0.7$, $\rho_a = 1.29$ kg/m^3, $\rho_f = 720$ kg/

m^3

Determine (a) the air-fuel ratio supplied by the carburettor if the pressure drop causing air flow is 0.14 bar neglecting nozzle lip and (b) the air-fuel ratio when nozzle lip is considered.

10.4 A simple carburettor fitted on a SI engine consumes 6.4 kg/h. The nozzle lip = 3 mm, d_f = 1.27 mm, C_{da} = 0.8 and C_{df} = 0.6. The atmospheric air pressure and temperature are 1.013 bar and 15.5°C and ρ_f = 760 kg/m^3. The A:F ratio supplied by the carburettor = 15:1. Determine (a) the venturi diameter, and (b) the drop in venturi pressure.

[**Ans.** 21.5 mm, 37 cm of H_2O]

10.5 A single jet carburettor is to be

designed to give an A:F ratio of 15:1 and fuel supply of 24.5 kg/h. The atmospheric air pressure and temperature are 1.013 bar and 288 K. Calculate (a) the venturi diameter if the air velocity is limited to 97.5 m/s, and (b) the fuel nozzle diameter.

Take $C_{da} = 0.84$ and $C_{df} = 0.66$.

[**Ans.** 33.8 mm, 2 mm]

10.6 A four-stroke, eight-cylinder CI engine develops 500 kW at 1200 rpm and consumes 0.175 kg/kWh of fuel whose specific gravity is 0.89. Determine the diameter of a single orifice injector if the injection pressure is 160 bar and combustion chamber pressure is 40 bar. The period of injection is 30° crank rotation. Take C_{df}

= 0.9.

10.7 A four-stroke, six-cylinder oil engine, 115 mm diameter and 140 mm stroke operates with A:F = 16:1. The suction condition in the engine are 1 bar and 300 K. The volumetric efficiency of the engine is 76%. Determine the maximum amount of fuel supplied per hour to the engine running at 1500 rpm. The injection pressure is 125 bar and combustion chamber pressure is 40 bar. The fuel is supplied for 20° rotation of the crank. Calculate the diameter of the orifice assuming each nozzle has single orifice. Take ρ_f = 760 kg/m^3 and C_{df} = 0.64.

[**Ans.** 21.7 kg/h, 1.76 mm]

10.8 A six-cylinder, four-stroke engine,

8 cm bore and 12 cm stroke runs at 3600 rpm. The volumetric efficiency of the engine is 0.8. If the maximum head causing the flow is limited to 11.765 cm of mercury, find the throat diameter of the venturi required. Find the diameter of the nozzle orifice if the desired A:F ratio is 15:1. Assume $C_{da} = 0.9$, $C_{df} = 0.7$, $\rho_a = 1.3$ kg/m^3, $\rho_f = 720$ kg/m^3.

[**Ans.** 28 mm, 1.69 mm]

10.9 The throat diameter of a carburettor is 8 cm and nozzle diameter is 5.5 mm. The nozzle lip is 6 mm. The pressure difference causing the flow is 0.1 bar. Find (*a*) A:F ratio supplied by the carburettor neglecting nozzle lip. (*b*) A: F ratio considering nozzle lip, and (*c*) and minimum velocity of air required to

start the fuel flow. Neglect air compressibility. Take $C_{da} = 0.85$, $C_{df} = 0.7$, $\rho_a = 1.2$ kg/m^3, $\rho_f = 750$ kg/m^3.

[**Ans.** 10.2:1, 10.28:1, 8.57 m/s]

10.10 The A:F ratio of a mixture supplied to an engine by a carburettor is 15:1. The fuel consumption of the engine is 7.5 kg/h. The diameter of the venturi is 2.2 cm. Find the diameter of fuel nozzle if the lip of nozzle is 4 mm. Assume $\rho_f = 750$ kg/m^3, $C_{da} = 0.82$, $C_{df} = 0.7$ atmospheric pressure 1.013 bar and temperature = 25°C. Neglect compressibility effect of air.

[**Ans.** 1.245 mm]

10.11 A simple carburettor is designed to supply 6 kg of air and 0.45 kg of fuel per minute to a four stroke single-

cylinder petrol engine. The ambient air is at 1.013 bar and 27°C. (a) Calculate the throat diameter of the choke (venturi) when the velocity of air is limited to 92 m/s. Take $\rho_f = 740$ kg/m^3 and velocity coefficient = 0.8 (b) If the pressure drop near the fuel nozzle is 75% of that at the venturi, calculate fuel nozzle diameter. Take $C_{df} = 0.6$.

[**Ans.** 35.2 mm, 23.4 mm]

10.12 Determine the size of fuel orifice to give air-fuel ratio of 12:1. The diameter of venturi throat is 3.5 cm and vacuum at the venturi is 6.9 cm of Hg. The pressure and temperature of atmospheric air are 1.013 bar and 25°C. The nozzle lip is 5 mm. Assume C_{da} = 0.9, $C_{df} = 0.7$, $\rho_f = 760$ kg/m^3. Consider

the compressibility of air.

[**Ans.** 0.63 mm]

10.13 An engine having a simple single jet carburettor consumes 6 kg/h of fuel. The level of fuel in the float chamber is 3 mm below the top of the jet when engine is not running. Ambient conditions are 76 cm of Hg pressure and 20°C temperature. The jet diameter is 1.3 mm and its coefficient of discharge is 0.60. The discharge coefficient of air is 0.84 and A/F ratio is 15. Determine the critical air velocity and the throat diameter. Express the pressure depression in cm of water. Take density of fuel = 700 kg/m3.

[**Ans.** 5.847 m/s, 19 mm, 32 cm of water]

10.14 A six-cylinder, four-stroke oil

engine develops 200 kW at 1200 rpm and consumes 0.3 kg/kWh. Determine the diameter of a single orifice injector if the injection pressure is 200 bar and combustion chamber pressure is 40 bar. The injection is carried out for 30° rotation of the crank. Take $\rho_f = 900$ kg/m^3 and $C_{df} = 0.7$. Each nozzle on a cylinder is provided with a single orifice.

[**Ans.** 1.005 mm]

10.15 A four-stroke, six-cylinder oil engine operates on A:F = 20:1. The diameter and stroke of the cylinder are 10 cm and 14 cm, respectively. The volumetric efficiency is 80%. The condition of air at the beginning of compression are 1 bar and 300 K.

Determine (*a*) the maximum amount of fuel that can be injected in each cylinder per cycle. (*b*) If the speed of the engine is 1500 rpm, injection pressure is 150 bar, air pressure during fuel injection is 40 bar and fuel injection is carried out for 20° of crank rotation, determine the diameter of the fuel orifice assuming only one orifice being used. Take ρ_f = 760 kg/m^3 and C_{df} = 0.67.

[**Ans.** 0.023 kg/s, 0.575 mm]

ANSWERS TO MULTIPLE-CHOICE QUESTIONS

1. c
2. c
3. a
4. a
5. d
6. c
7. b
8. a
9. a
10. d
11. a
12. b
13. b
14. d

15. b
16. a
17. b
18. d
19. a
20. d
21. b
22. d
23. b
24. d
25. d
26. d
27. a
28. a
29. b
30. d
31. a
32. a
33. d
34. b
35. a
36. b
37. a
38. d
39. c
40. a
41. a
42. c
43. b
44. c
45. c
46. b
47. d
48. b
49. c
50. c
51. c
52. c
53. b
54. b
55. a
56. a
57. d

58. a
59. b
60. c
61. a
62. c
63. b
64. c
65. b
66. d
67. b
68. b
69. a
70. d
71. c
72. a

Chapter 11

Performance of Internal Combustion Engines

11.1 ❒ PERFORMANCE PARAMETERS

The performance of an engine is an indication of the degree of success with which the conversion of chemical energy contained in the fuel is done into useful mechanical work. The degree of success is compared on the basis of following parameters:

1. Specific fuel consumption (SFC)
2. Brake mean effective pressure (BMEP)
3. Specific power output (SP)
4. Specific weight (SW)
5. Exhaust smoke and other emissions

However, in the evaluation of engine performance, the following performance

parameters are chosen:

1. Power and mechanical efficiency
2. Mean effective pressure and torque
3. Specific output
4. Fuel-air ratio
5. Volumetric efficiency
6. Specific fuel consumption
7. Thermal efficiency and heat balance
8. Exhaust smoke and other emissions
9. Specific weight

Indicated power,

where p_{im} = indicated mean effective pressure, Pa

L = Length of stroke, *m*

A = Area of stroke, m^2

N = rotational speed, rpm

n = number of strokes per revolution

= 2 for four-stroke engine

= 1 for two-stroke engine

where p_{bm} = brake mean effective pressure, Pa

Specific output: It is defined as the brake output per unit of piston displacement.

Volumetric efficiency,

$$= v_a / v_s$$

Brake specific fuel consumption,

where m_f = mass of fuel consumed, kg

t = time in minutes

Indicated specific fuel consumption,

where LCV = lower calorific value of fuel, kJ/kg

where η_a = air standard efficiency

Exhaust smoke and other emissions: Oxides in nitrogen, unburned hydrocarbons, etc.

11.2 ❒ BASIC ENGINE MEASUREMENTS

The basic measurements required to be carried out to evaluate the performance

of an engine are as follows:

1. Speed
2. Fuel consumption
3. Air consumption
4. Smoke density
5. Brake power
6. Indicated power and friction power
7. Heat loss to cooling water
8. Heat going to exhaust
9. Exhaust gas analysis

1. **Speed Measurement:** The angular speed of engine (crankshaft) may be measured by a tachometer. Tachometers may be classified into the following categories:
    1. Mechanical tachometers
    2. Electrical tachometers

    They can further be classified as contact type and non-contact type. Contact type tachometers are of the following types:

    1. **Mechanical type:** Revolution counter and timer, slipping clutch tachometers, and centrifugalfree tachometers.
    2. **Electrical type:** Drag cup tachometer, electromagnetic tachogenerator. Non-contact type tachometers are photoelectric tachometer, variable reluctance tachometer, stroboscope, and capacitance pick up tachometer.
2. **Fuel consumption measurement:** The fuel consumption of an engine is measured by determining the volume flow in a given time interval and multiplying by the specific gravity of the fuel. Another method is to measure the time required for consumption of a given mass of fuel. Accurate methods for fuel measurement are as follows:
    1. Two glass-chambers method
    2. Rotameter
    3. Positive displacement meter

3. **Measurement of air consumption:** Accurate measurements for air consumption can be carried out by the following methods:
    1. Orifice chamber method
    2. Viscous-flow air meter
4. **Measurement of exhaust smoke:** A smoke meter is a sort density measuring device and may be used to measure exhaust smoke.
5. **Measurement of heat carried away by exhaust gases:** An exhaust gas calorimeter is commonly used for the measurement of heat carried by exhaust gases.

Mass of air supplied per kg of fuel,

Heat carried away by exhaust gas per kg of fuel,

where $(m_a + 1)$ = mass of exhaust gases formed per kg of fuel supplied

c_{pg} = specific heat of exhaust gases, kJ/kgk

T_{ge} = temperature of exhaust gases coming out from engine, °C

T_a = temperature of ambient air, °C

6. **Measurement of heat carried away by cooling water:** The heat carried away by cooling water is generally measured by measuring the water flow rate through the cooling jacket and the rise in temperature of water during flow through the engine. The inlet and outlet temperatures of water are measured by thermometers inserted in the pockets provided at the inlet and outlet from the engine. The quantity of water flowing is measured by collecting the water in a bucket for a specified period.

Heat carried away by cooling water,

where m_w = mass of water collected /min

c_{pw} = specific heat of water, kJ/kg °C

T_{wi} = inlet temperature of water, °C

T_{w0} = outlet temperature of water, °C

11.3 ❐ HEAT BALANCE SHEET

A heat balance sheet is an account of heat supplied and heat utilised during engine trial.

1. Heat supplied to the engine,

$$Q_s = \dot{m}_f \times \text{LCV}$$

where $\dot{m}_f$ = mass of fuel supplied per hour.

LCV lower calorific value of fuel, kJ/kg

2. Heat utilised in various ways:
    1. Heat equivalent of *BP*,

    $$Q_1 = 3600 \times BP \text{ in kW kJ/h}$$

    2. Heat carried away by cooling water,

    $$Q_2 = \dot{m}_w \cdot c_{pw} (T_{w0} - T_{wi}) \text{ kJ/h}$$

    3. Heat carried away by exhaust gases,

    $$Q_3 = \dot{m}_g \cdot c_{pg} (T_{ge} - T_a) \text{ kJ/h}$$

    4. Heat unaccounted for,

    $$Q_4 = Q_s - (Q_1 + Q_2 + Q_3)$$

11.4 ❐ WILLAN'S LINE METHOD

It is a method to determine the frictional power of an engine. In this method, gross fuel consumption *v*'s brake power at a constant speed is plotted and the graph is extrapolated back to zero fuel consumption as shown in Fig. 11.1. The point where this graph cuts the BP axis is an indication of the frictional power of the engine at that speed. The test is applicable to only compression ignition engines.

Figure 11.1 *Willan's line method*

11.5 ❒ MORSE TEST

This is a method to determine the indicated power of a cylinder in a multi-cylinder engine. In this method, the engine is first run at the required speed and the output is measured. One

cylinder is then cut out by short circuiting the spark plug or by disconnecting the injector, as the case may be. The output is measured by keeping the speed constant at its original value. The difference in the output is a measure of the indicated power of the cutout cylinder. Thus for each cylinder the IP is obtain and is added together to find the total IP of the engine.

IP of n cylinders, $IP_n = BP_n + FP_n$

IP of $(n - 1)$ cylinders $IP_{n-1} = BP_{n-1} + FP_n$

IP of nth cylinder, $(IP)_{nth} = BP_n - BP_{n-1} = IP_n - IP_{n-1}$

Total IP of engine, $IP_n = \sum (IP)_{nth}$

11.6 ❐ PERFORMANCE OF SI ENGINES

The performance of an engine is generally given by a heat balance sheet. The main components of a heat balance sheet are (i) heat equivalent to the effective (brake) work of the engine, (ii) heat rejected to the cooling medium, (iii) heat carried away from the engine with the exhaust gases, and (iv) unaccounted losses. The unaccounted losses include the radiation losses from the various parts of the engine and heat loss due to incomplete combustion. The friction loss is not shown as a separate item as it ultimately reappears as heat in cooling water, exhaust, and radiation.

Figure 11.2 shows the heat balance sheet for a petrol engine run at full

throttle over its speed range. In SI engines, the loss due to incomplete combustion included in unaccounted losses can be rather high. For a rich mixture (A /F ratio 12.5:13) it could be 20%. Figure 11.3 shows the heat balance of an uncontrolled SI engine at different loads.

Figure 11.2 *Heat balance diagram for a typical SI engine*

Figure 11.3 *Uncontrolled SI engine*

Figure 11.4 *Efficiency and specific fuel consumption vs speed for a SI engine at full throttle*

Figure 11.4 shows the brake thermal efficiency, indicated thermal efficiency, mechanical efficiency, and specific fuel consumption for the above SI engine.

Figures 11.5 (a) and (b) show the indicated power (IP), brake power (BP),

and friction power (FP) (by difference), brake torque, brake mean effective pressure, and brake specific fuel consumption of a high compression ratio (r) automotive SI engine at full or wide open throttle (WOT).

The following conclusions can be drawn from the above figures:

1. At full throttle, the brake thermal efficiency at various speeds varies from 20%–27%, maximum efficiency being at the middle speed range.
2. The percentage heat rejected to a coolant is more at lower speed (>> 35%) and reduces at higher speeds (>> 25%). Considerably more heat is carried by the exhaust at higher speeds.
3. Torque and mean effective pressure (MEP) do not strongly depend on the speed of the engine but depend on volumetric efficiency and friction losses. Maximum torque position corresponds with the maximum air charge or maximum volumetric efficiency position.

   Torque and MEP curves peak at about half that of the brake power.

Figure 11.5 *Variable speed test of automotive SI engine at full throttle (CR = 9): (a) Power, mep vs engine speed, (b)* η_{mech}*, SFC vs engine speed*

Note: If size (displacement) of the engine were to

be double, torque would also double, but MEP is a ‘specific’ torque—a variable independent of the size of the engine.

4. High brake power arises from high speed. In the speed range before the maximum brake power is obtained, doubling the speed doubles the power.
5. At low engine speed, the frictional power is relatively low and brake power is nearly as large as indicated power. As engine speed increases, however, friction power increases at continuously greater rate and therefore, brake power reaches a peak and starts reducing even though indicated power is increasing. At engine speeds above the usual operating range, friction power increases very rapidly. At these higher speeds, indicated power will reach a maximum and then fall off. At some point, indicated power and friction power will be equal, and brake power will then drop to zero.

11.6.1 Performance of SI Engine at Constant Speed and Variable Load

The performance of an SI engine at constant speed and variable load is different from the performance at full throttle and variable speed. Figure 11.6 shows the heat balance of an SI engine at constant speed and variable load. The load is varied by altering the throttle and the speed is kept constant by resetting

the dynamometer.

Figure 11.6 *Heat balance vs load for a SI engine*

Closing the throttle reduces the pressure inside the cylinders but the temperature is affected very little because the air / fuel ratio is substantially constant, and the gas temperatures throughout the cycle are high. This results in high loss to coolant at low engine loads.

At low loads, the efficiency is about 10%, rising to about 25% at full load. The loss to the coolant is about 60% at low loads and 30% at full load. The exhaust temperature rises very slowly with load and as the mass flow rate of exhaust gas is reduced because the mass flow rate of fuel into the engine is reduced, the percentage loss to exhaust

remains nearly constant (about 21% at low loads to 24% at full load). Percentage loss to radiation increases from about 7% at low loads to 20% at full load.

11.7 ❐ PERFORMANCE OF CI ENGINES

The performance of a CI engine at constant speed and variable load is shown in Fig. 11.7. As the efficiency of the CI engine is more than the SI engine, the total losses are less. The coolant loss is more at low loads and radiation and other losses are greater at high loads. The BMEP, BP, and torque directly increase with load, as shown in Figs. 11.8(a) and (b). Unlike the SI engine, the BP and BMEP are continuously rising curves and are

limited only by the smoke. The exhaust temperature is also nearly proportional to the load. The lowest BSFC and hence the maximum efficiency occurs at about 80% of the full load.

Figure 11.7 *Heat balance diagram for a typical CI engine*

Figure 11.8 *IP, BP, IMEP, BMEP, and SFC for a CI engine*

Figure 11.9 *Performance curves of a diesel engine*

Figure 11.9 shows the performance curves of a variable speed GM 7850 cc four cycle V-6 Toro-flow diesel engine. The maximum torque value is at about 70% of maximum speed compared to about 50% in the SI engine. In addition, the BSFC is low through most of the speed range for the diesel engine and is better than the SI engine.

11.8 ❐ PERFORMANCE MAPS

For critical analysis, the performance of an IC engine under all conditions of load and speed is shown by a performance map. Figure 11.10 shows the performance map of an automotive SI engine and Fig. 11.11 shows the performance map of a four-stroke pre-chamber CI engine. Figure 11.11 also includes a typical curve of BMEP versus piston speed for level road operation in high gear. Note that these maps can be used for comparing engines of different sizes as performance parameters have been generalised by converting rpm into piston speed and brake power per time of piston area.

Figure 11.10 *Form of performance map for a SI engine*

Figure 11.11 *Form of performance map for a diesel engine*

Generally, all engines show a region of the lowest specific fuel consumption (highest efficiency) at a relatively low piston speed with a relatively high BMEP.

SI Engine: *Constant speed line*: Increased BSFC is obtained by moving upward along the constant speed line because of mixture enrichment at high load which offsets an increase in mechanical efficiency. Moving to lower BMEP, the BSFC increases because of the reduced mechanical efficiency (IMEP decreases, whereas FMEP remains constant).

Constant BMEP line: Moving from the region of the highest efficiency along a line of constant BMEP, the BSFC

increases due to increased friction at higher piston speeds. Moving to the left towards lower piston speed, although friction MEP decreases, indicated efficiency falls off owing to poor fuel distribution and increased relative heat losses.

CI Engine: In the CI engine, the BSFC increases at high loads owing to increased fuel waste (smoke) associated with high fuel-air ratios. At lower load, BSFC increases due to decrease in mechanical efficiency (same in the SI engine).

As speed is reduced from the point of the best economy along a line of constant BMEP, the product of mechanical and indicated thermal

efficiency appears to remain about constant down to the lowest operating speed. The reduction in FMEP with speed is apparently balanced by a reduction in indicated thermal efficiency due to poor spray characteristics at very low speed.

An interesting feature of performance curves is that they show that power at maximum economy is about half of the maximum power.

Example 11.1

A six-cylinder, four-stroke spark-ignition engine of 10 cm × 2 cm (bore × stroke) with a compression ratio of 6 is tested at 4800 rpm on a

dynamometer of arm 55 cm. During a 10 minutes test, the dynamometer reads 45 kg and the engine consumed 5 kg of petrol of calorific value 45 MJ/kg. The carburettor receives air at 9°C and 1 bar at the rate of 10 kg /min. Calculate (a) the brake power, (b) the brake mean effective pressure, (c) the brake specific fuel consumption, (d) the brake specific air consumption, (e) the brake thermal efficiency, and (f) the air-fuel-ratio.

Solution

Given that $i = 6$, $n = 2$, $\Delta = 10$ cm, $L = 12$ cm, $r = 6$, $N = 4800$ rpm, $l = 55$ cm, $t = 10$ min, $w = 45$ kg, $m_f = 5$ kg, CV = 45 MJ/kg, $\dot{m}_a = 10$ kg /

min

1. Brake power,

$$T = w \times g \times l = 45 \times 9.81 \times 55 \times 10^{-2} = 242.79 \text{ N-m}$$

2. BP = p_{bm} [(LA)N/n] × Number of cylinders

$$p_{bm} = 5.39 \text{ bar}$$

3.
4. $\dot{m}_a$ = 10 kg /min or 600 kg /h
5.
6.

Example 11.2

A sharp-edged circular orifice of diameter 3.8 cm and coefficient of discharge as 0.6 is used to measure air-consumption of a four-stroke petrol engine. Pressure drop through the orifice is 145 mm of water and barometer reads 75.5 cm of Hg. The compression ratio of the engine is 6 and the piston displacement volume is 2000 cm^3.

The temperature of air is taken to be 26°C. At 2600 rpm, the engine brake power recorded is 29.5 kW. The fuel consumption is 0.14 kg / min and the calorific value of fuel used is 43960 kJ/kg. Determine (a) the volumetric efficiency, (b) the air-fuel ratio, (c) the brake mean effective pressure, (d) the brake thermal efficiency, (e) the air standard efficiency, and (f) the relative efficiency.

Solution

Given that $d = 3.8$ cm, $C_d = 0.6$, $h_w = 14.5$ cm of water, $p_a = 75.5$ cm of Hg, $r = 6$, $V_s = 2000\ \text{cm}^3$, $N = 2600$ rpm, BP = 29.5 kW, $\dot{m}_f = 0.14$ kg / min, CV = 43960 kJ/kg

Density of air at atmospheric conditions,

Head in *m* of air =

Velocity of air passing through orifice,

$$c_a = [2gH]^{0.5} = [2 \times 9.81 \times 124.43]^{0.5} = 49.41 \text{ m/s}$$

Volume of air passing through orifice,

$$= 0.033622 \text{ m}^3/\text{s or } 2.0173 \text{ m}^3/\text{min}$$

1. Volumetric efficiency
2. Mass flow rate of air,

$$\dot{m}_a = V_a \rho_a = 2.0173 \times 1.1653 = 2.35 \text{ kg/min}$$

3. BMEP =
4. Brake thermal efficiency
5. Air standard efficiency,
6. Relative efficiency =

Example 11.3

A four-stroke SI engine has six single-acting cylinders of 7.5 cm bore and 9 cm stroke. The engine is coupled to a brake having a torque arm radius of 38 m. At 3300 rpm, with all cylinders operating the net brake load is 324 N. When each cylinder, in turn, is rendered in operative, the average net brake load produced at the same speed of the remaining five cylinders is 245 N. Estimate the indicated mean effective pressure engine.

Solution

BP when all cylinders are working

=

= 42.55 kW

BP when each cylinder is cut-off in turn

IP of cylinder cut off = 42.55 – 32.17 = 10.38 kW

Total IP of engine = 6 × 10.38 = 62.3 kW

Indicated mean effective pressure

Example 11.4

A six-cylinder SI engine operates in a four-stroke cycle. The bore of

each cylinder is 70 mm and the stroke 100 mm. The clearance volume per cylinder is 67 cm^3. At a speed of 3960 rpm the fuel consumption is 19.5 kg /h and the torque developed is 140 N-m. Calculate (a) brake power, (b) brake mean effective pressure, (c) the brake thermal efficiency if LCV of fuel is 44000 kJ/kg, and (d) the relative efficiency on brake power basis. Assume $\gamma = 1.4$ for air.

Solution

1. Brake power
2. Brake mean effective pressure
3. Brake thermal efficiency
4. Swept volume per cylinder,

   Compression ratio,

   Air standard efficiency,

Relative efficiency,

11.9 ❒ MEASUREMENT OF AIR CONSUMPTION BY AIR-BOX METHOD

The arrangement to measure the consumption of air by air-box method is shown in Fig. 11.12. It consists of an air-tight box fitted with a sharp-edged orifice of known coefficient of discharge. Due to the suction of engine, there is a pressure depression in the box which causes the flow through orifice for obtaining a steady flow. The pressure difference causing the flow through the orifice is measured with the help of a water manometer.

Let

A_0 = area of orifice, m^2

h_w = head of water causing flow, cm

C_d = coefficient of discharge for orifice

d_0 = diameter of orifice, cm

ρ_a = density of air, kg /m^3

Head of air,

Velocity of air through the orifice,

Figure 11.12 *Measurement of air by air box method*

Volume of air passing through the orifice,

Mass flow rate of air passing through the orifice,

Density of air,

where h_w is in cm, p_a in bar and T_a in K

Mass of air supplied per kg of fuel used,

where N = % of N_2 by volume in exhaust gases

C = % of carbon in fuel

C_1, C_2 = % of CO_2 and CO by volume in exhaust gases.

11.10 ❐ MEASUREMENT OF BRAKE POWER

Brake power is measured with the following ways:

1. **Rope brake dynamometer:** A rope brake dynamometer is shown in Fig. 11.13. A rope is wound around the brake drum, whose one end is connected to the spring balance *S* suspended from overhead and the other end carries the load *W*. In this

arrangement, the whole power developed by the engine is absorbed by the friction produced at the rim of the brake drum. The rim of the brake drum is generally water-cooled to absorb the heat generated due to rubbing action of rope on rim.

Let

W = dead weight on the rope, N

S = spring pull, N

D = outer diameter of brake drum, m

d_r = diameter of rope, m

N = speed of engine, rpm

Net load acting on brake drum = $(W - S)$ N

Effective radius of brake drum,

Frictional torque rating on brake drum,

$$T_f = (W - S) \times R \text{ Nm}$$

Figure 11.13 *Rope brake dynamometer*

2. **Prony brake:** The arrangement of this braking system is shown in Fig. 11.14. It consists of brake shoes which are clamped on the rim of the brake drum by means of bolts. The pressure on the rim is adjusted with the help of nuts and springs. A load lever extends from top of the brake and a load carrier is attached to the end of the load lever. The weights kept on this load carrier are balanced by the reaction torque in the shoes. The load lever is kept horizontal to keep its length constant.

Figure 11.14 *Prony-brake arrangement*

where

W = load on load carrier, N

L = distances from centre of shaft to load, m.

Example 11.5

A four-cylinder engine running at 1200 rpm gave 18.6 kW brake power. The average torque when one cylinder was cut off was 105 N.m. Determine the indicated thermal efficiency if the calorific value of the fuel is 42,000 kJ/kg and the engine uses 0.34 kg of petrol per kWh of brake power.

Solution

Given that $i = 4$, $N = 1200$ rpm, $(\mathrm{BP})_n = 18.6$ kW, $T_3 = 105$ N.m, $CV = 42{,}000$ kJ/kg

BSFC = 0.34 kg /kWh

$$(\mathrm{BP})_n - (\mathrm{BP})_{n-1} = (\mathrm{IP})_n - (\mathrm{IP})_{n-1} = (\mathrm{IP})_1$$

$\therefore (\mathrm{IP})_1 = 18.6 - 13.2 = 5.4$ kW

Indicated power of 4 cylinders,

$$(\mathrm{IP})_n = i \times (\mathrm{IP})_1 = 4 \times 5.4 = 21.6 \text{ kW}$$

Fuel consumption,

Indicated thermal efficiency,

Example 11.6

A single cylinder four-stroke oil engine works on a diesel cycle. The following readings correspond to full load conditions:

Area of indicator diagram = 3 cm^2,
Length of the diagram = 5 cm,
Spring constant = 12 bar/cm^2. cm,
Speed of the engine = 500 rpm,
Load on the brake = 400 N, Spring reading = 50 N

Diameter of brake drum = 120 cm,
Fuel consumption = 2.75 kg /h,
Calorific value of fuel = 42000 kJ/kg, Diameter of cylinder = 16 cm,
Stroke length of piston = 20 cm.

Determine (a) the friction power, (b) the mechanical efficiency, (c) the brake thermal efficiency, and (d) the brake mean effective pressure.

Solution

Given that $n = 2$, $A_i = 3$ cm^2, $L_i = 5$ cm, $C = 12$ bar/cm^2. cm, $N = 500$ rpm, $W = 400$ N, $S = 50$ N, $D_b = 120$ cm, $\dot{m}_f = 2.75$ kg /h, $CV = 42{,}000$ kJ/kg, $d = 16$ cm, $L = 20$ cm

1. Frictional power,

$$FP = IP - BP = 12.06 - 10.99 = 1.07 \text{ kW}$$

2. Mechanical efficiency
3. Brake thermal efficiency
4. Brake mean effective pressure

Example 11.7

The following data are known for a

four-cylinder four-stroke petrol engine: cylinder dimensions: 11 cm bore and 13 cm stroke, engine speed: 2250 rpm, brake power: 50 kW, friction power: 15 kW, fuel consumption rate: 10.5 kg /h, calorific value of fuel: 50,000 kJ/kg, air inhalation rate: 300 kg /h, and ambient condition: 15°C, 1.03 bar. Estimate (a) brake mean effective pressure, (b) volumetric efficiency, (c) brake thermal efficiency, and (d) mechanical efficiency.

[IES, 2008]

Solution

Given that D = 11 cm, L = 13 cm, N = 2250 rpm, BP = 50 kW, FP = 15 kW, $\dot{m}_f$ = 10.5 kg /h. CV = 50,000

kJ/kg, $\dot{m}_a$ = 300 kg /h, t_a = 15°C, p_a = 1.03 bar, i = 4

1. or

2. Volumetric efficiency

   Swept mass of air per second

3. Brake thermal efficiency,
4. Indicated power,

$$IP = BP + FP = 50 + 15 = 65 \text{ kW}$$

   Mechanical efficiency

Example 11.8

The following data relates to a four-cylinder, four-stroke petrol engine:

Diameter of piston = 80 mm, length of stroke = 120 mm

Clearance volume = 100×10^3 mm^3

Fuel supply = 4.8 kg /h. calorific value = 44100 kJ/kg

When the Morse test was performed the following data were obtained:

BP with all cylinders working = 14.5 kW

BP with cylinder 1 cut-off = 9.8 kW

BP with cylinder 2 cut-off = 10.3 kW

BP with cylinder 3 cut-off = 10.14 kW

BP with cylinder 4 cut-off = 10 kW

Find the IP of the engine and

calculate indicated thermal efficiency, brake thermal efficiency, and relative efficiency.

Solution

Given that $i = 4$, $n = 2$, $d = 80$ mm, $L = 120$ mm, $V_c = 100 \times 10^3$ mm^3, $\dot{m}_f = 4.8$ kg /k, $CV = 44100$ kJ/kg, $(BP)_n = 14.5$ kW, $(BP)_1 = 9.8$ kW, $(BP)_2 = 10.3$ kW, $(BP)_3 = 10.14$ kW, $(BP)_4 = 10$ kW

$$(IP)_1 = (BP)_n - (BP)_{n-1}$$

$$\therefore \quad (IP)_1 = 14.5 - 9.8 = 4.7 \text{ kW}$$

$$(IP)_2 = 14.5 - 10.3 = 4.2 \text{ kW}$$
$$(IP)_3 = 14.5 - 10.14 = 4.36 \text{ kW}$$
$$(IP)_4 = 14.5 - 10 = 4.5 \text{ kW}$$

Indicated power,

Indicated thermal efficiency,

Brake thermal efficiency,

Swept volume,

Compression ratio,

Air standard efficiency,

Relative efficiency,

Example 11.9

Find the air-fuel ratio of a four-stroke, single-cylinder, air-cooled engine with a fuel consumption time for 10 cc as 20.4 s and air

consumption time for 0.1 m^3 as 16.3 s. The load is 17 kg at the speed of 3000 rpm. Find the brake specific fuel consumption and brake thermal efficiency. Assume the density of air as 1.175 kg /m^3 and specific gravity of fuel to be 0.7. The lower heating value of fuel is 43MJ/kg and dynamometer constant is 5000.

Solution

Given that W = 17 kg, N = 3000 rpm, ρ_a = 1.175 kg /m^3, s_f = 0.7, LCV = 43 MJ/kg, C = 5000

$\dot{m}_f = \dot{V}_f \rho_f = \dot{V}_f \times s_f \times \rho_w = 0.49 \times 10^{-6} \times 0.7 \times 10^3 = 0.343 \times 10^{-3}$ kg / s

$\dot{m}_a = \dot{V}_a \rho_a = 6.135 \times 10^{-3} \times 1.175 = 7.208 \times 10^{-3}$ kg /s

A /F ratio =

Brake power,

Brake specific fuel consumption,

Brake thermal efficiency,

Example 11.10

The following data refers to a test on a single cylinder oil engine working on four-stroke cycle:

Diameter of brake wheel = 60 cm; rope diameter = 3 cm; dead load is

25 kg, spring balance reading = 5 kg, and the engine is running at 400 rev/min. The indicator diagram has area = 4 cm^2, length = 6 cm, and spring stiffness is 12 bar/cm. The fuel consumption is 0.23 kg /kWh and the fuel used has a calorific value 43963.5 kJ/kg. Taking cylinder bore 10 cm and piston stroke 15 cm, calculate the brake power, indicated power, mechanical and indicated thermal efficiencies of the engine.

Solution

Given that $D_b = 0.6$ m, $d_r = 0.03$ m, $W = 25 \times 9.81 = 245.25$ N, $S = 5 \times 9.81 = 49.05$ N, $N = 400$ rpm, $n = 2$, $a_2 = 4$ cm^2, $l_i = 6$ cm, $k = 12$ bar/cm,

$\dot{m}_f = 0.23$ kg /kWh, CV = 43963.5 kJ/kg, $D = 10$ cm, $L = 15$ cm

Indicated mean effective pressure,

Indicated power,

Mechanical efficiency,

Total fuel consumption, $\dot{m} = \text{BP} \times \dot{m}_f$

$$= 2.589 \times 0.23 = 0.5955 \text{ kg /h}$$

Indicated thermal efficiency,

Example 11.11

A four-stroke, single cylinder petrol

engine mounted on a motor cycle was put to load test. The load measured on dynamometer was 30 kg with drum diameter and speed, respectively, at 900 mm and 2000 rpm. The engine consumed 0.15 kg of fuel in one minute, the calorific value of fuel being 43.5 MJ/kg. The fuel supply to the engine was stopped and was driven by a motor which needed 5 kW of power to keep it running at the same speed, the efficiency of the motor being 80%. The engine cylinder bore and stroke are respectively at 150 mm and 200 mm. Calculate (a) the brake power, (b) the indicated power, (c) the mechanical efficiency, (d) the brake thermal efficiency, (e) the

indicated thermal efficiency, (f) the brake mean effective pressure and (vii) the indicated mean effective pressure.

[IES, 2012]

Solution

Given that four-stroke single cylinder petrol engine, $W = 30$ kg, $D_d = 900$ mm, $N = 2000$ rpm, $\dot{m}_f =$ 0.15 kg /min, CV = 43.5 MJ/kg, P_m = 5 kW, $\eta_m = 80\%$, $D = 150$ mm, L = 200 mm, $n = 2$.

1. Torque applied,

   Break power,

2. Indicated power,

$$IP = BP + FP$$

   Frictional power,

Indicated power, IP = 27.737 + 6.25 = 33.987 kW

3. Mechanical efficiency,
4. Brake thermal efficiency,
5. Indicated thermal efficiency,
6. BMEP, P_{bm} = 4.7 bar
7.

Example 11.12

During the trial of a single oil engine, cylinder diameter 20 cm, stroke 28 cm, working on the two-stroke cycle, and firing every cycle, the following observations were made:

Duration of trial = 1 h, total fuel used = 4.22 kg, calorific value of fuel = 44670 kJ/kg, proportion of hydrogen in fuel = 15%, total number of revolutions = 21,000,

mean effective pressure = 2.74 bar, net brake load applied to a drum of 100 cm diameter = 600 N, total mass of cooling water circulated = 495 kg, temperature of cooling water: inlet 13°C, outlet 38°C, air used = 135 kg, temperature of air in test room = 20°C, temperature of exhaust gases = 370°C. Assume c_p of gases = 1.005 kJ/kg.K, c_p of steam = 2.093 kJ/kg.K at atmospheric pressure. Calculate the thermal efficiency and draw up the heat balance sheet.

[IES, 1997]

Solution

Given that D = 20 cm, L = 28 cm, n = 1, t = 1 h, m_f = 4.22 kg, CV =

44670 kJ/kg, H_2 = 15%, N_t = 21000 rev, p_m = 2.74 bar, W = 600 N, d = 100 cm, m_w = 495 kg, t_{wi} = 13°C, t_{wo} = 38°C, m_e = 13.5 kg, t_r = 20°C, t_g = 370°C, c_{pg} = 1.005 kJ/kg. K, c_{ps} = 2.093 kJ/kg.K

Torque, $T = W \times d/2 = 600 \times 0.5 = 300$ N.m

Brake power,

Indicated power,

Thermal efficiency,

Swept volume,

Heat input,

Heat equivalent of BP, Q_1 = BP × 60 = 10.996 × 60 = 659.76 kJ/min

Heat lost to cooling water,

Heat carried away by exhaust gases:

Mass of flue gases =

Steam in exhaust gases

Mass of dry exhaust gases $\dot{m}_g$ = 2.32 – 0.095 = 2.225 kg /min

Heat in steam in exhaust gases, $Q_3 = \dot{m}_s\,[h_g + c_{ps}\,(t_g - t_{sat}) - h_{fw}]$ [$\therefore t_{sup} = t_g$] at p_{atm} = 1.013 bar

$$= 0.095\,[2676 + 2.093\,(370 - 100) - 4.187 \times 20]\ [\because h_{fw} = c_{pw} \times t_r]$$
$$= 300 \text{ kJ/min}$$

Heat in dry exhaust gases = $\dot{m}_g \times c_{pg}$ $(t_g - t_r)$

$$= 2.225 \times 1.005 \times (370 - 20) = 782.6 \text{ kJ/min}$$

Unaccounted heat loss = 3141.79 – [659.76 + 862.125 + 300 + 782.6]

$$= 537.305 \text{ kJ/min}$$

Heat Balance Sheet:

Example 11.13

During a trial of a single cylinder, a four-stroke diesel engine, the following observations were recorded:

Bore = 340 mm, stroke = 440 mm, rpm = 400, area of indicator

diagram = 465 mm^2, length of diagram = 60 mm, spring constant = 0.6 bar/mm, load on hydraulic dynamometer = 950 N, dynamometer constant = 7460, fuel used = 10.6 kg /h, calorific value of fuel (CV) = 49500 kJ/kg, cooling water circulated = 25 kg /min, rise in temp. of cooling water = 25°C, mass analysis of fuel: carbon = 84%, hydrogen = 15%, incombustible = 1%, volume analysis of exhaust gas: carbon dioxide = 9%, oxygen = 10%, temperature of exhaust gases = 400°C, specific heat of exhaust gas = 1.05 kJ/kg°C, partial pressure of steam in exhaust gas = 0.030 bar, specific heat of superheated steam =

2.1 kJ/kg°C, saturation temp. of steam at 0.030 bar = 24.1°C. Draw up heat balance sheet on minute basis.

[IAS, 2011]

Solution

Given that $n = 2$, $D = 0.34$ m, $L = 0.44$ m, $N = 400$ rpm, $A_i = 465$ mm^2, $L_d = 60$ mm, $k = 0.6$ bar/mm, $W = 950$ N, $C = 7460$, CV = 49500 kJ/kg, $\dot{m}_f = 10.6$ kg /h, $\dot{m}_w = 25$ kg / min, $\Delta t_w = 25°C$, $C = 84\%$, $H_2 = 15\%$, incombustible = 1%, $CO_2 = 9\%$, $O_2 = 10\%$, $t_{eg} = 400°C$, $c_{pg} = 1.05$ kJ/kg /°C, $p_s = 0.030$ bar, $c_{ps} = 2.1$ kJ/kg°C, t_s at 0.030 bar = 24.1°C

Brake power,

Indicated mean effective pressure,

p_{im} =

Indicated power, IP =

= 61.92 kW

1.
2.
   1. Heat equivalent of BP, Q_b = BP × 60 = 50.938 × 60 = 3056.28 kJ/min
   2. Heat lost to cooling water, $Q_w = \dot{m}_w - c_{pw} \cdot \Delta t_w$ = 25 × 4.187 × 25 = 2616.87 kJ/min

Percentage of N_2 in exhaust gas = 100 – (9 + 10) = 81%

Mass of air supplied per kg of fuel,

Mass of exhaust gases formed per kg of fuel = 1 + 22.91 = 23.91 kg

Mass of exhaust gases formed per minute =

Mass of steam formed and carried with the exhaust gases per minute due to the combustion of hydrogen in the fuel,

Mass of dry exhaust gases formed per minute, $\dot{m}_g = 4.224 - 0.2385 = 3.9855$ kg /min

3. Heat carried away by exhaust gases per minute, $Q_g = \dot{m}_g \, c_{pg} \, (t_g - t_a)$

Let ambient air temperature, $t_a = 27°C$

$$Q_g = 3.9855 \times 1.05 \, (400 - 27) = 1560.92 \text{ kJ/min}$$

4. Heat carried away by steam with exhaust gases per minute,

$$Q_s = \dot{m}_s \, [h_g + c_{ps} \, (T_{sup} - 100)]$$

At $p = 0.030$ bar, $t_s = 24.10°C$, $h_g = 2545.5$ kJ/kg

$$Q_s = 0.2385 \, [2545.5 + 2.1 \, (400 - 100)] = 757.36 \text{ kJ/min}$$

3. Heat unaccounted for $= Q_i - (Q_b + Q_w + Q_g + Q_s)$

$$= 8745 - (3056.28 + 2616.87 + 1560.92 \times 757.36) = 753.57 \text{ kJ/min}$$

Heat Balance Sheet:

Example 11.14

During a test on a two-stroke engine on full load, the following

observations were recorded:

Speed =350 rpm, net brake load = 590 N, mean effective pressure = 2.8 bar, fuel oil consumption = 4.3 kg /h, cooling water required = 500 kg /h, rise in cooling water temperature = 25°C, air used per kg of fuel = 33 kg, room temperature = 25°C, exhaust gas temperature = 400°C, cylinder diameter = 220 mm, stroke length = 280 mm, effective brake diameter = 1 m, C.V. of fuel oil = 43900 kJ/kg, proportion of hydrogen in fuel = 15%, mean specific heat of exhaust gases = 1.0 kJ/kg-K, specific heat of steam = 2.09 kJ/kg-K

Calculate the following:

1. Indicated power
2. Brake power
3. Draw the heat balance sheet on the basis of kJ/min

[IAS, 2010]

Solution

Given that N = 350 rpm, W_b = 590 N, p_{mi} = 2.8 bar, $\dot{m}_f$ = 4.3 kg /h, $(\Delta t)_w$ = 25°C, $\dot{m}_w$ = 500 kg /h, m_a / m_f = 33, t_0 = 25°C, t_g = 400°C, D = 220 mm, L = 280 mm, d_b = 1 m, CV = 43900 kJ/kg, H_2 = 15%, c_{pg} = 1.0 kJ/kg.K, c_{ps} = 2.09 kJ/kg.K

1. Indicated power.
2. Brake power,
3. Heat balance sheet:

Heat input =

Heat equivalent of BP = 10.81 × 60 = 648.6 kJ/min

Heat lost to cooling water = $\dot{m}_w \times c_{pw} \times (\Delta t)_w$ =

Heat carried away by exhaust gases:

Mass of flue gases =

Steam in exhaust gases,

Mass of dry exhaust gases, $\dot{m}_g$ = 2.437 – 0.09675 = 2.340 kg /min

$$h_f = c_{pw}\,(100 - t_0)$$

Heat in steam in exhaust gases = $\dot{m}_s\,[h_f + h_{fg} + c_{ps}\,(t_g - 100)]$

$$= 0.09675\,[4.187\,(100 - 25) + 2256.9 + 2.09\,(400 - 100)]$$
$$= 309.4 \text{ kJ/min}$$

Heat in dry exhaust gases = $\dot{m}_g \times c_{pg}\,(t_g - 100)$ = 2.340 × 1.0 (400 – 100) = 702 kJ/min

Unaccounted heat loss = 3146.2 – (648.6 + 872.3 + 309.4 + 702) = 613.9 kJ/min

Example 11.15

A four-stroke petrol engine develops 30 kW at 2600 rpm. The compression ratio of the engine is 8 and its fuel consumption is 8.4 kg /h

with calorific value of 44 MJ/kg. The air consumption of the engine as measured by means of a sharp-edged orifice is 2 m^3 per min. If the piston displacement volume is 2 litres, calculate (a) the volumetric efficiency, (b) the air-fuel ratio, (c) the brake mean effective pressure, (d) the brake thermal efficiency, and (e) the relative efficiency.

The ambient temperature air can be taken as 27°C, R for air as 287 J/kgK, and g = 1.4. The barometer reads 755 mm of mercury.

[IAS, 2006]

Solution

Given that $n = 2$, BP = 30 kW, $N =$

2600 rpm, $r = 8$, $\dot{m}_f = 8.4$ kg /h, CV = 44 MJ/kg, $\dot{V}_a = 2 m^3/min$, $V_s = 2$ litres = 2×10^{-3} m^3, $T_0 = 27 + 273 = 300$ K, $R = 287$ J/kg.K, $\gamma = 1.4$, $p_b =$ 755 mm of Hg

1. Swept volume per minute,

   Volumetric efficiency

2.

   $$\dot{m}_a = \dot{V}_a \times 60 \times \rho_a = 2 \times 60 \times 1.169 = 140.28 \text{ kg /h}$$

3. Brake power BP =

   or

   or $p_s = 0.6923$ MPa

4. Brake thermal efficiency = 0.2922 or 29.22%

   $$\eta_a = 1 - (1/r^{\gamma - 1}) = 1 - 1/(0.8)^{0.4} = 0.5647 \text{ or } 56.47\%$$

5. Relative efficiency

Example 11.16

A six-cylinder, four-stroke petrol engine has a swept volume of 3.0 litres with a compression ratio of

9.5. Brake output torque is 205 N-m at 3600 rpm. Air enters at 10^5 N/m^2 and 60°C. The mechanical efficiency of the engine is 85% and air-fuel ratio is 15:1. The heating value of fuel is 44,000 kJ/kg and the combustion efficiency is 97%. Calculate (a) the rate of fuel flow, (b) the brake thermal efficiency, (c) the indicated thermal efficiency, (d) the volumetric efficiency, and (e) the brake specific fuel consumption.

[IAS, 2005]

Solution

Given that $i = 6$, $n = 2$, $V_s = 3$ litre = 3×10^{-3} m^3, $r = 9.5$, $T_b = 205$ N.m, $N = 3600$ rpm, $p = 105$ N/m^2, $T = 60 + 273 = 333$ K, $\eta_{mech} = 85\%$, A /F =

15, CV = 44000 kJ/kg, $\eta_{comb} = 97\%$

Brake power,

Indicated power,

Air standard efficiency,

Density of air,

Air volume flow rate,

Mass of air taken in per second = $0.09 \times 1.046 = 0.09414$ kg /s

1. Mass flow rate of fuel for the engine,
2. Brake thermal efficiency,

   = 0.2885 or 28.85%
3. Indicated thermal efficiency,
4. Relative efficiency,
5.

Example 11.17

The air flow to a four cylinder, four-stroke engine is measured by means of a 4.5 cm diameter orifice, having $C_d = 0.65$. During a test the following data was recorded:

Bore = 10 cm, stroke = 15 cm, engine speed = 1000 rpm, brake torque = 135 Nm, fuel consumption = 5.0 kg /h, CV_{fuel} = 42600 kJ/kg, head across orifice = 6 cm of water.

Ambient temperature and pressure are 300 K and 1.0 bar, respectively.

Calculate (a) the brake thermal efficiency, (b) the brake mean effective pressure, and (c) the volumetric efficiency.

[IAS, 2004]

Solution

Given that $i = 4$, $n = 2$, $d_0 = 4.5$ cm, $C_d = 0.65$, $D = 10$ cm, $L = 15$ cm, $N = 1000$ rpm, $T_b = 135$ Nm, $\dot{m}_f = 5$ kg /h, CV = 42600 kJ/kg, $h = 6$ cm of H_2O, $T_0 = 300$ K, $p_0 = 1$ bar, $R = 287$ J/kg.K

1. Brake power,

   Brake thermal efficiency,

2. Brake mean effective pressure,

3. $\Delta p = \rho g h = 10^3 \times 9.81 \times 6 \times 10^{-2} = 588.6$ N/m^2

   Air inhaled per second,

   Swept volume/s,

   Volumetric efficiency,

Example 11.18

An eight-cylinder automobile engine of 80 mm diameter and 90 mm stroke with a compression ratio of 7, is tested at 4000 rpm on a dynamometer of 600 mm arm length. During a 10-minute test period at a dynamometer scale reading of 450 N, 4.8 kg of gasoline having a calorific value of 45000 kJ/kg was burnt and air at 27°C and 1.0 bar was supplied to the carburettor at the rate of 6.6 kg /min. Find (a) the brake power delivered, (b) the brake mean effective pressure, (c) the brake specific fuel consumption, (d) the brake thermal efficiency, (e) the volumetric efficiency, and (f) the air fuel ratio.

[IAS, 2003]

Solution

Given that $i = 8$, $D = 80$ mm, $L = 90$ mm, $r = 7$, $N = 4000$ rpm, $l = 600$ mm, $t = 10$ min, $W = 450$ N, $m_f =$ 4.8 kg, CV = 45000 kJ/kg, $T_1 = 27 + 273 = 300$ K, $p_1 = 1$ bar, $\dot{m}_a = 6.6$ kg /min

Density of air,

1. Torque on the dynamometer, $T = Wl = 450 \times 0.6 = 270$ N.m

   Brake power,

2. or

   or $p_{bm} = 9.375$ bar

3. Brake specific fuel consumption
4. Brake thermal efficiency,
5. Swept volume,

   Swept volume/s,

   Air inhaled: $p_1\dot{V} = \dot{m}_a RT_1$

Volumetric efficiency,

6.

Example 11.19

A six-cylinder, four-stroke petrol engine, with a bore of 120 mm and stroke of 180 mm under test, is supplied petrol of composition: C = 82% and H_2 = 18% by mass. The Orsat gas analysis indicated that CO_2 = 12%, O_2 = 4% and and N_2 = 84% by volume. Determine (a) the air-fuel ratio and (b) the percentage of excess air.

Calculate the volumetric efficiency of engine based on intake conditions

when the mass flow rate of petrol is 32 kg /min at 1600 rpm. Intake condition are 1 bar and 17°C. Consider the density of petrol vapour to be 3.5 times that of air at same temperature and pressure. Air contains 23% oxygen by mass.

[IAS, 2003]

Solution

Given that $i = 6$, $n = 2$, $D = 120$ mm, $L = 180$ mm, $C = 82\%$, $H_2 = 18\%$, $CO_2 = 12\%$, $O_2 = 4\%$, $N_2 = 84\%$, $\dot{m}_f = 32$ kg /h, $N = 1600$ rpm, $p_0 = 1$ bar, $T_0 = 17 + 273 = 290$ K, $\rho_f / \rho_a = 3.5$

Minimum mass of air required for complete combustion

Mass of air supplied,

Excess air supplied = 17.394 – 15.768 = 1.626 kg /kg of fuel

Percentage of excess air =

Actual A/F ratio = 17.394:1 kg air/ kg of fuel

Mass flow rate of petrol = 32 kg /h

Swept volume per second =

Volume of air,

Volume of petrol vapour,

Total volume = 14.477 + 2.913 = 17.39 m3/kg fuel

Mixture aspirated per minute =

Volumetric efficiency,

Example 11.20

Two identical petrol engines having the following specifications are used in vehicles:

Engine 1: Swept volume = 3300 cc. normally aspirated, BMEP = 9.3 bar, rpm ≃ 4500, compression ratio = 8.2, efficiency ratio = 0.5, mechanical efficiency ≃ 0.9, mass of the engine ≃ 200 kg

Engine 2: Supercharged, swept volume = 3300 cc, BMEP ≃ 12.0

bar, rpm = 4500, compression ratio = 5.5, efficiency ratio = 0.5, mechanical efficiency = 0.92, mass of the engine ≃ 220 kg

If both the engines are supplied with just adequate quantity of petrol for the test run, determine the duration of test run so that the specific mass per kW of brake power is same for both the engines. Calorific value of petrol = 44000 kJ/kg.

Assume both the engines operate on four stroke cycle.

[IAS, 2010]

Solution

Engine 1

$V_s = 3300$ cc or 33×10^{-4} m^3, $p_{bm} =$ 9.3 bar, $N = 4500$ rpm, $n = 2$, $r =$ 8.2, $\eta_r = 0.5$, $\eta_{mech} = 0.9$, $M_1 = 200$ kg, LCV = 4400 kJ/kg

Brake power,

Indicated power, IP = BP/η_{mech} = 115/0.9 = 127.8 kW

Air standard efficiency,

Efficiency ratio,

or $\quad \eta_{\text{th}} = 0.5 \times 0.569 = 0.2845$

Again, $\quad \text{IP} = m_{f\,1} \times \text{CV} \times (\eta_{\text{th}})_i$

or $\quad 127.8 = m_{f\,1} \times 44000 \times 0.2865$

or $\quad m_{f\,1} = 10.96 \times 10^{-3}$ kg /s

$$= 10.96 \times 10^{-3} \times t \times 3600 = 39.46 \times t \text{ kg/h}$$

Specific mass of engine per kW of

Engine 2:

$V_s = 3300$ cc or 33×10^{-4} m^3, $p_{bm} =$ 12 bar, $N = 450$ rpm, $n = 2$, $r = 5.5$, $\eta_r = 0.5$, $\eta_{mech} = 0.92$, $M_2 = 220$ kg

Air standard efficiency,

$$(\eta_{\text{th}})_i = 0.494 \times 0.5 = 0.247$$
$$\text{IP} = m_{f2} \times \text{CV} \times (\eta_{\text{th}})_i$$

or $\quad 161.3 = m_{f2} \times 44000 \times 0.247$

or $\quad m_{f2} = 14.84 \times 10^{-3}$ kg/s

$$= 14.84 \times 10^{-3} \times t \times 3600 = 53.43 \times t \text{ kg/h}$$

Specific mass of engine per kW of

Now

or $\quad 1.2904\,(39.46 \times t + 200) = 53.43 \times t + 220$

or $\quad 38.08 = 2.511 \times t$

or $\quad t = 15.1$ hours

Example 11.21

The following data relates to a test trial of a single-cylinder four-stroke engine:

Cylinder dia. = 24 cm, stroke length = 48 cm, compression ratio = 5.9,

number of explosions/minute = 77, gas used/min at 771 mm of mercury and 15°C = 0.172 m^3, lower calorific value of gas at NTP = 49350 kJ/m^3, mean effective pressure from indicator card = 7.5 bar, weight of Jacket cooling water/min = 11 kg, temperature rise of cooling water = 34.2°C, specific heat of water = 4.2 kJ/kg°C.

Net brake load applied at brake wheel having an effective circumference of 3.86 m is 1260 N at average speed of 227 rpm.

Estimate (a) the mechanical efficiency, (b) the indicated thermal efficiency and (c) the efficiency ratio, and draw a heat balance sheet

for the engine assuming that exhaust gases carry away 24% of heat.

[IAS, 2000]

Solution

Given that D = 24 cm, L = 48 cm, r = 5.9, W_{bn} = 1260 N, = 0.6143 m, N_m = 227 rpm, N_e = 77/min, $\dot{V}_g$ = 0.217 m^3/min at 15°C and 771 mm of Hg, LCV = 49350 kJ/m3 at NTP, p_{mi} = 7.5 bar, $\dot{m}_w$ = 11 kg /min, $(\Delta t)_w$ = 34.2°C, c_{pw} = 4.2 kJ/kg.°C

1. Brake torque, $T_b = W_{bn} \times r_b = 1260 \times 0.6143 = 774$ N.m

   Brake power,

   Indicated power,

   Mechanical efficiency,

2. Indicated thermal efficiency,

   Volume of gas at NTP = 0.20873 m^3/min

Indicated thermal efficiency

3. Air standard efficiency, η_a =

Heat balance sheet:

Heat input = $\dot{V}_0$ × LCV = 0.20873 × 49350 = 10300.82 kJ/min

Heat equivalent of BP = 18.4 × 60 = 1104 kJ/min

Heat lost to cooling water = $\dot{m}_w \times c_{pw} \times (\Delta t)_w$

$$= 11 \times 4.2 \times 34.2 = 1580.04 \text{ kJ/min}$$

Heat lost to exhaust gases = 0.24 × 10,300.82 = 2472.20 kJ/s

Heat unaccounted for = 10,300.82 – (1104 + 1580.04 + 2472.20) = 5144.58 kJ/min

11.11 ❐ SUPERCHARGING OF IC ENGINES

The power output of an engine depends on the amount of air inducted into the

cylinder per unit time, the degree of utilisation of this air, and the thermal efficiency of the engine. The amount of air inducted per unit time can be increased by increasing the engine speed or by increasing the density of air at intake. The volumetric efficiency decreases as the speed is increased. The method of increasing the inlet air density is called *supercharging*. It is generally used to increase the power output of the engine. It is done by supplying air at a pressure higher than the pressure at which the engine naturally aspirates air from the atmosphere by using a pressure boosting device called a *supercharger*.

11.11.1 Thermodynamic Cycle

The p-v diagram for an ideal Otto-cycle supercharged engine is shown in Fig. 11.15(a). The pressure p_1 represents the supercharging pressure and p_6 is the exhaust pressure. Area 8-6-7-0-1-8 represents the work done by the supercharger in supplying air at a pressure p_1, whereas the area 1-2-3-4-1 is the output of the engine. Area 0-1-6-7-0 represents the gain in work during the gas exchange process due to supercharging. Thus, a part of the supercharger work is recovered. However, the area 1-6-8-1 cannot be recovered and represents a loss of work. This loss of work causes the ideal thermal efficiency of the supercharged engine to decrease with an increase in supercharging pressure.

Figure 11.15(b) shows an ideal dual combustion cycle supercharged engine. The pressure p_1 represents the supercharging pressure and $p_6 = p_7$, is the exhaust pressure. The engine is supercharged by the compressor. Area 9-10-1-11-9 represents the work done on the supercharger in supplying air at pressure p_1. Thus, for the engine, 8-1 represents induction process, 1-2 as the compression process, 2-3-4 as the heat addition process, 4-5 as the expansion process, 5-6 as the blow down to atmosphere or heat rejection process and 6-7 as the exhaust process.

Figure 11.15 *p-v diagram for supercharged engine: (a) Ideal Otto cycle, (b) Dual combustion cycle*

Work done my mass *m* by the supercharged engine,

W = Area (1-2-3-4-5-1) + Area (8-1-6-7-8) — Area (9-10-1-11-9).

Thus, a part of the supercharger work is recovered; however, the curve equivalent to Area (9-10-1-11-9) — Area (8-1-6-7-8) is not recoverable and represents loss of work. This loss of work causes the ideal thermal efficiency of supercharged engine to decrease with an increase in supercharging pressure.

11.11.2 Supercharging of SI Engines

As far as SI engines are concerned, supercharging is employed only for aircraft and racing car engines. This is because the increase in supercharging

pressure increases the tendency to detonate and pre-ignite.

Apart from increasing the volumetric efficiency of the engine, supercharging results in an increase in the intake temperature and pressure of the engine. These reduce the ignition delay and increase the flame speed and results in greater tendency to detonate or pre-ignite. For this reason, the supercharged SI engines use a lower compression ratio which results in lower thermal efficiency. The fuel consumption is also greater than naturally aspirated engines.

Due to its poor fuel economy, supercharging of petrol engines is not very popular and is used only when more power is needed or when more

power is need to compensate altitude loss.

11.11.3 Supercharging of CI Engines

Supercharging of CI engines does not result in any combustion problems. Increase in pressure and temperature of the intake air reduces ignition delay and hence the rate of pressure rise results in a better, quieter, and smoother combustion. This allows the use of poor quality fuel. The increase in intake air temperature reduces volumetric and thermal efficiencies but the increase in density due to pressure compensates for this and intercooling is not required except for highly supercharged engines. It is possible to use lower fuel-air ratios in a supercharged engine, resulting in

lower temperature and reduced smoke from the engine. This results in an increased life of the engine. *Thus, a CI engine is more suitable for supercharging*.

The apparatus used for increasing air density is known as a supercharger. A supercharger is an air compressor which may be positive displacement, reciprocating (piston-cylinder), rotary type (roots blower), or rotodynamic compressors (centrifugal or axial flow type). The centrifugal compressor is widely used as a supercharger for IC engines. The function of a supercharger is either to produce more power from an engine of a given cylinder size, or to compensate the power loss at high

altitudes due to rarefied atmosphere.

11.11.4 Effects of Supercharging

The effect of supercharging on the power and efficiency is shown in Fig. 11.16.

1. **Power output:** Supercharging produces more power because the supercharger supplies air at a pressure and density higher than atmospheric with has the effect of increasing volumetric efficiency.
2. **Mechanical efficiency:** The mechanical efficiency of a supercharged engine is higher than that of one not supercharged.
3. **Fuel consumption:** It provides better mixing of fuel and air, which results in a specific reduction of fuel consumption and the thermal efficiency increases.

Figure 11.16 *Effect of supercharging ratio on power and efficiency*

11.11.5 Objectives of Supercharging

The objectives of supercharging are as follows:

1. To overcome the effect of high altitudes, as in the case of aircrafts and stationary engines in mountains.
2. To reduce the weight of an engine per kW or power developed, as in the case of racing cars.
3. To reduce the size of the engine to fit into a limited space, as in

the case of locomotives or marine engines.

4. To increase the power output of an existing engine to meet greater power demands.

11.11.6 Configurations of a Supercharger

1. **Compressor Coupled of Engine Shaft:** The compressor is operated directly by the engine with set-up gearing to increase the speed of centrifugal compressor. A part of the engine output is used to drive the supercharger and the net output is calculated by deducing this power from the gross power of the engine.
2. **Turbo-charger:** The energy of the exhaust gases from the engine is used to develop power in a turbine which directly runs the supercharger. There is no mechanical connection between the engine and the supercharger.
3. **Direct Coupling between the Engine, Compressor, and Turbine:** The advantage of this arrangement is that when the turbine output is insufficient to run the compressor, additional power required is taken from the engine. Additional power from the turbine can also be fed to the engine.
4. **Compressor geared with Engine and Free turbine:** The engine drives the compressor but the energy in the exhaust is utilised to develop the power from a separate turbine.

The various configurations are shown in Fig. 11.17(a) to (d).

Figure 11.17 *Arrangements for supercharging: (a) Mechanical supercharging, (b) Turbocharging, (c) Engine-driven compressor and turbocharger, (d) Engine-driven compressor and free turbine*

11.11.7 Supercharging of Single Cylinder Engines

Supercharging of single cylinder

engines is not carried out because the thermal efficiency drops to zero when the ratio of supercharger is about 6. Further, the power is maximum when the ratio of the supercharger is nearly 2, 5, and then drops to zero at a ratio of about 6.

Example 11.22

A four-stroke diesel engine of 3000 cc capacity develops 14 kW per m^3 of free air induced per minute. When running at 3500 rev/min, it has a volumetric efficiency of 85% referred to free air-conditions of 1.013 bar and 27°C. It is proposed to boost the power of the engine by supercharging by a blower (driven

mechanically from the engine) of pressure ratio 1.7 and isentropic efficiency of 80%. Assuming that at the end of induction, the cylinders contain a volume of charge equal to the swept volume, at the pressure and temperature of the delivery from the blower, estimate the increase in brake power to be expected from the engine. Take overall mechanical efficiency as 80%, γ for air = 1.4, R = 0.287 kJ/kg K.

[IES, 2009]

Solution

The schematic arrangement is shown in Fig. 11.18(a).

Swept volume of engine per minute,

Unsupercharged inducted volume

$$= V_s \times \eta_{vol} = 5.25 \times 0.85 = 4.4625 \text{ m}^3/\text{min}$$

The actual and isentropic compression processes of supercharger are shown in Fig. 11.18(b).

Isentropic efficiency of supercharger.

Figure 11.18 *Supercharger: (a) Schematic arrangement, (b) Processes on T-s diagram*

The supercharger delivers 5.25 m^3/min at $p_2 = 1.013 \times 1.7 = 1.72$ bar

and 361.4 K to the engine. This volume referred to 1.013 bar and 300 K is,

Increase in inducted volume = $V - V_s$ = 7.41 – 4.4625 = 2.95 m^3/min

As indicated power (I.P.) is directly proportional to induced volume, therefore, increase in I.P. due to increase in inducted volume of air

$$= 14 \times 2.95 = 41.3 \text{ kW}$$

Increase in IP due to increase in inducted air pressure because of supercharger

Total increase in IP because of supercharger

$$= 41.3 + 6.186 = 47.486 \text{ kW}$$

Increase in BP = Increase in IP × η_{mech}

$$= 47.486 \times 0.8 = 38 \text{ kW}$$

Power required to run the supercharger.

$$P_{\text{sup}} = \dot{m}_a \cdot c_{pa} (T_2' - T_1)$$

where $\dot{m}_a$ = mass of air delivered by the supercharger per second.

$$P_{\text{sup}} = 0.145 \times 1.005 \times (361.4 - 300) = 8.95 \text{ kW}$$

Net increase in BP = 38 – 8.95 = 29.05 kW

Percentage increase in

11.12 ❒ SI ENGINE EMISSIONS

Spark ignition engine emissions are divided into three categories as exhaust emission, evaporative emission, and crank case emission.

Figure 11.19 *Emissions from SI engines*

The major constituents which contribute to air pollution are CO, NO_x, and hydrocarbons (HC) coming from SI engine exhaust. The percentages of different constituents coming out from the above three mentioed sources are shown in Fig. 11.19.

The relative amounts depend on the engine design and operating conditions but are of order, $NO_x \rightarrow$ 500 to 1000 ppm (20 g /kg of fuel), CO $\rightarrow$ 1 to 2% (200 g /kg of fuel) and HC $\rightarrow$ 3000 ppm(25 g /kg of fuel). Fuel evaporation

from the fuel tank and the carburettor exists even after engine shut down and these are unburned HCs. However, in most modern engines, these non-exhaust unburned HCs are effectively controlled by returning the blow-by gases from the crank case to the engine intake system by venting the fuel tank and through a vapour-absorbing carbon cannister which is purged as some parts of the engine intake air during normal engine operation.

The other constituents include SO_2 and lead compounds. Petrol rarely contains sulphur; therefore, SO_2 is not a pollutant from the SI engine exhaust. Petrol contains lead in small percentages but its effect is more serious on human

health.

The processes by which pollutants form within the cylinder in a conventional SI engine are qualitatively illustrated in Fig. 11.19. It shows the formation of pollutants during four strokes of the cycle. NO forms throughout the high temperature burned gases behind the flame through chemical reactions. NO formation rate increases with an increase in gas temperature. As the burned gases cool, during expansion stroke, the reactions involving NO freeze and leave NO concentrations far in excess of levels corresponding to equilibrium temperature at exhaust conditions.

CO also forms during combustion

process with lean A:F mixtures, and there is sufficient O_2 to burn all the carbon in the fuel to CO_2. However, in high temperature products even with lean mixtures, there is sufficient CO in exhaust because of dissociation of CO_2. Later, in expansion stroke, the CO oxidation process also freezes as the gas temperature falls.

The unburned hydrocarbon emission comes from different sources. During compression and combustion, the increasing cylinder pressure forces some of the gases in the cylinder into crevices connected to combustion chamber, the volumes between the piston rings and cylinder wall are the largest of these. Most of this gas entering into crevices is

unburned air fuel mixture escaped from primary combustion zone. This happens because the flame cannot enter these narrow crevices. The gas which leaves these crevices later in the expansion and exhaust processes is one source of unburned HC emissions. The combustion chamber walls are another possible source. A quench layer containing unburned and partially burned A:F mixture is left at the wall when the flame dies as it approaches the wall. This unburned HC in this layer (0.1 mm) burns rapidly if the combustion chamber walls are clean. The next source of HC is the thin layer of lubricating oil on cylinder wall, and the piston which absorbs HC before and after combustion. A final source of HC

in engines is incomplete combustion due to bulk quenching of the flame in that fraction of engine cycle where the combustion is especially slow. This unburned HC near the cylinder wall is exhausted during exhaust stroke as the piston pushes the gases out.

11.12.1 Exhaust Emissions

The major exhaust emissions are as follows:

1. Unburnt hydrocarbons, (HC)
2. Oxides of carbon, (CO and CO_2)
3. Oxides of nitrogen, (NO and NO_2)
4. Oxides of sulphur, (SO_2 and SO_3)
5. Particulates
6. Soot and smoke

The various exhaust emissions from petrol engine are as follows:

1. **Unburnt hydrocarbons (HC):** The causes for the emissions of HC are incomplete combustion, crevice volumes and flow in crevices, leakage past the exhaust valve, valve overlap,

deposits on walls, and oil on combustion chamber walls.

The reasons for incomplete combustion are improper mixing of air and fuel, and flame quenching at the walls of the cylinder. Low load and idle conditions increase HC.

2. **Oxides of Carbon (CO and CO_2):** Carbon monoxide is generated with a rich fuel-air ratio mixture. This happens during starting and accelerating under load. Poor mixing, local rich regions, and incomplete combustion are the sources of CO emissions.
3. **Oxides of Nitrogen (NO_x):** NO_x are created mostly from nitrogen in the air and fuel blends.In addition to temperature, the formation of NO_x depends on pressure, A/F ratio, combustion duration, and location of spark plug.
4. **Oxides of Sulphur (SO_x):** These are generated due to the presence of sulphur in the fuel. SO_2 and SO_3 react with water to give rise to H_2SO_3 and H_2SO_4, which causes acid rain.

The variation of emissions from a petrol engine with A/F ratio is shown in Fig. 11.20.

Figure 11.20 *Variation of emissions with A/F*

11.12.2 Evaporative Emission

As mentioned earlier, there are two main sources of evaporative emissions—the fuel tank and the carburettor. The

main factors governing the tank emissions are fuel volatility and ambient temperature but the tank design and location can also influence the emissions as location affects the temperature. Insulation of the fuel tank and vapour collection systems have all been explored with a view to reduce the tank emissions.

Carburettor emission may be divided into two categories as running losses and parking losses. Most internally vented carburettors have an external vent which opens at idle throttle position. The existing pressure forces prevent outflow of vapours to the atmosphere. Internally vented carburettor may enrich the mixture

which, in turn, increases exhaust emission. Carburettor losses are significant only during hot conditions when the vehicle is in operation. Fuel volatility also affects the carburettor emissions.

11.12.3 Crankcase Emission

It consists of engine blow by-gases and crank case lubricant fumes. From the point of view of pollution, blow-by gases are the most important. The blow-by is the phenomenon of leakage past the piston from the cylinder to the crankcase because of pressure difference. The blow of HC emissions is about 20% of the total HC emission from the engine. This is further increased to 30% if the piston-rings are

worn.

11.12.4 Lead Emission

Lead emissions come only from SI engines. The lead is present in the fuel as lead tetraethyl or tetramethyl, to control the self-ignition tendency of fuel-air mixtures that is responsible for knock.

11.13 ❐ CONTROL OF EMISSIONS IN SI ENGINE

An emission control programme aims at reducing the concentration of CO, HC, and NO. The main approaches adopted are as follows:

1. **Engine design modification:** The following steps reduce the exhaust emission by engine design modification:
    1. Use leaner air-fuel ratio
    2. Retard ignition timing
    3. Avoid flame quenching time by reducing the surface to volume ratio of the combustion chamber
    4. Lower compression ratio
    5. Reduce valve overlap
    6. Modify the induction system by using high velocity

    carburettors or multi-chock carburettors
2. **Exhaust gas oxidation:** The devices used to reduce HC/CO emissions are as follows:
    1. Use of after-burner
    2. Use of exhaust manifold reactor
    3. Use of catalytic converter
3. **Fuel modification:** Air-fuel ratios leaner than stoichiometric result in almost insignificant amount of CO and reduce HC with reduced specific fuel consumption.
4. **Blow-by control:** The crankcase blow-by control is the recirculation of the vapours back to the intake air cleaner.

To reduce atmospheric pollution, two different approaches are followed:

1. Reduction of formation of pollutants in the emission by redesigning the engine system, fuel system, cooling system, and ignition system.
2. Destroying the pollutants after these have been formed.

In petrol engines, the main pollutants which are objectionable and are to be reduced are HC, CO, and NO_x. The methods used are as follows:

1. **Modifications in the engine design:** Engine modifications improve emission quality. A few parameters which improve an emission are as follows:
    1. **Combustion chamber configuration:** Modification in the combustion chamber as reducing surface/volume ratio can reduce quenching zone and reduce HC emission. This can also be achieved by reducing dead space around piston ring.

2. **Lower compression ratio:** Lower compression ratio also reduces the quenching area and thus reduces HC emission. Lower compression ratio also reduces NO_x emission due to lower maximum temperature. However, lowering compression ratio reduces thermal efficiency and increases fuel consumption. By using petrol of lower octane number, it is possible to phase the lead out of petrol, that is, use of unleaded petrol.
3. **Induction system:** The supply of designed A:F ratio mixture to multi-cylinder engine is always difficult under all operating conditions of load and power. This can be achieved by proper designing of induction system or using high velocity or multiple-choke carburettors.
4. **Ignition timing:** Retarding spark ignition allows increased time for the fuel to burn. Retarding the spark reduces NO_x formation by decreasing NO_x emission. It also reduces HC emission by causing higher exhaust temperature. However, retarding the ignition results in loss of power and consumption of fuel. The controls are designed to retard the spark timing during idling and provide normal spark advance during acceleration.
5. **Reduced valve overlap:** Increased overlap carries fresh mixture with the exhaust and increases emission level. This can be avoided by reducing the valve overlap.

2. **Modifying the fuel used:** To reduce the pollution from the exhaust of these engines, the emission of olefins should be obviated as far as possible. For this, we can change the fuel itself. LPG and CNG be used instead of gasoline as they produce less pollution than present petrol engines.
3. **Exhaust gas treatment:** *Exhaust Gas Oxidation*—The exhaust gases coming out of exhaust manifold are treated to reduce HC and CO emission. A few devices are discussed below.
    1. **Use of after-burner:** An after-burner is a burner where air is supplied to the exhaust gases and the mixture is burned with the help of an ignition system. The HC and CO which are formed in the engine combustion chamber because of inadequate O_2 and inadequate time to burn are further burned by providing air in a separate box, known as an after-burner. The after-burner is located close to the

exhaust manifold with an intention that the temperature of the exhaust should not fall. The oxidation of HC in the after-burner depends on the temperature of the exhaust and the mixing provided in the after-burner. Air injection does nothing to NO_x emission. A simple arrangement of an after-burner is shown in Fig. 11.21.

The performance of this system was not satisfactory as combustion was not sustained during low HC emission.

Figure 11.21 *Typical after-burner*

2. **Exhaust manifold reactor:** This is a further development of the after-burner where high temperature exhaust gases and secondary air are mixed properly and burnt. Here HC carried with exhaust combines with O_2 and forms non-objectionable gases.

   There are different types of after-burners where heat losses are minimised and sufficient time and mixing of exhaust and secondary air are provided.

   A special after-burner designed by Du-Point, where the entry of exhaust gases is radial and air flow is peripheral, is shown in Fig. 11.22.

3. **Catalytic converters:** Catalytic oxidation of the exhausted HC and CO is accomplished by placing a common device called catalytic converter in the vehicle exhaust system. The catalytic converter is filled with catalytic material. Exhaust gas hydrocarbons and CO are oxidised while passing through the bed. The catalytic material itself does not

enter into the reaction but only promotes the oxidation process at a lower temperature. Usually, air compressor is used to supply additional oxygen necessary for complete oxidation of the exhaust gas stream.

A catalyst is an agent that aids or speeds a process or a chemical reaction without becoming a part of the reaction during the process—it is a sort of chemical middleman. In a modern car's emissions control system, the so-called three-way catalyst helps the three major evil elements of exhaust—HC, CO, and NO_x—react with oxygen and each other. The catalyst helps the HC and CO become non-poisonous CO_2 and water vapour, whereas the NO_x is converted into CO_2, nitrogen, and water vapour.

Figure 11.22 *A typical after-burner*

Figure 11.23 *Fuel-system evaporation loss control device: (a) Hot soak condition, (b) Purging condition*

4. **Evaporation Emission Control Device:** The purpose of this device is to collect all evaporative emissions (vapours) and recirculating them at a proper time.

   The device is shown in Fig. 11.23. It consists of an absorbent chamber, pressure balancing valve, and purge control valve. The absorbent chamber contains charcoal which can hold the hydrocarbon vapour before it escapes into the atmosphere. The fuel tank and carburettor float,

which are main sources of HC emission in the form of vapour, are directly connected to the absorbent chamber when the engine is turned off, that is, under hot soak-condition. This causes the petrol to boil from the carburettor float and a large amount of petrol vapour comes out. All these vapours during stopping or running the engine are absorbed in the absorber chamber.

When the absorber bed becomes saturated, the air coming out from the air-cleaner is passed through the absorber bed and the air with vapour is passed to the inlet manifold through the purge valve. Here, the seat of the pressure balancing valve is so located that there is direct pressure communication between the internal vent and the top of the carburettor float, maintaining the designed carburettor metering forces.

The operation of the purge control valve is controlled by the exhaust back pressure as shown in Fig. 11.23. The fuel supply is cut off under idling condition and the level of HC is reduced.

11.14 ❐ CRANK CASE EMISSION CONTROL

The basic principle of a crank case blow-by control system is the

recirculation of vapours back to the inlet manifold. Figure 11.24 shows a typical closed type known as a positive crank case ventilation (PCV) system. The gases escaping past the piston and entering into the crankcase are returned to the inlet manifold and then to the engine. During compression, the HC enters into the crankcase due to pressure difference between the engine cylinder and crank case (small vacuum) and the HC from the crank case is taken back into the intake manifold during the expansion stroke and back side of the piston compresses the gases in the crank cases.

11.15 ❐ CI ENGINE EMISSIONS

The diesel engine is used more than any

other type of engine for transportation, thermal power generation, and many other industrial and agricultural applications. The exhaust emissions from combustion in diesel engine are no different from those of combustion processes in petrol engine, the difference being only in the level of concentration of individual pollutants. The sample of a diesel exhaust may be free from smoke, odour, and HC or may be heavily smoke-laden, highly malodorous, and can have heavy concentration of unburned HC.

Figure 11.24 *Positive crank case ventilation system*

The pollutants from a diesel engine can be classified into two types as visible and invisible emissions. Visible

emission is the smoke which is objected more by the public. The invisible emissions include CO, HC, NO, SO_2, partially oxidised organics (as aldehydes and ketones), and odours. An unpleasant odour is also heavily objected by the public. Smoke and odour are not harmful to public health but are objectionable because of their unsightliness, unbearable smell, and possible reduction in visibility. Other invisible emissions mentioned above have similar effect on health as they are also emitted by petrol engines.

11.15.1 Effect of Engine Type on Diesel Emission

The type of the engine and the speed of the engine are two main factors which influence the exhaust emission from a

diesel engine. It has been observed that there is a significant difference in emission levels from different engines except the odour level.

The following observations can be summarised:

1. A two-stroke air-scavenged engine produces high HC and intermediate NO_x. The smoke level remains low at all load conditions.
2. A four-stroke medium speed engine has the lowest emissions of all constituents except high smoke intensity.
3. A four-stroke, high speed engine has high HC emissions.
4. A turbo-charged, four-stroke engine is notably low in HC but high in NO_x. The smoke level is also considerably low compared with other engines.

Formation of Smoke and Affecting Factors

Engine exhaust smoke is the result of incomplete combustion. Smoke from exhaust is a visible indicator of the combustion process within the engine. It is generated at any volume in the engine where the mixture is rich. The fuel-air

ratio greater than 1.5 and at pressures developed in diesel engine produces soot. Once soot is formed, it can burn if it finds sufficient O_2; otherwise it comes out with exhaust. It becomes visible if it is dense. The size of the soot particles affects the appearance of smoke. The soot particles agglomerate into bigger particles which have an objectionable darkening effect on diesel exhaust.

The turbo-charged engine emits less smoke with increasing fuel air ratio as the mixing is much better in this engine as well as sufficient O_2 is also available in the engine cylinder at all times.

Diesel Odour

It has been observed from the experiments that the products of partial

oxidation are the main cause of odour in diesel exhaust. This partial oxidation may due to a very lean mixture during idling or due to wall-quenching effect. The effect of fuel-air ratio and odour is shown in Fig. 11.25.

The members of the aldehyde family are considered responsible for the pungent odours of diesel exhaust. The aldehydes in the exhaust are found at maximum 30 ppm but it is observed that even 1 ppm can cause irritation to the nose and eyes. There is no standard method developed yet for measuring the odour. However, several odour producing components such as nepthaldehyde, *n*-butylbenzene, and so on, are given standard rating and trained personnel can give odour ratings

for the diesel exhaust sample by comparison.

The factors which affect the odour formation in the diesel engine are as follows:

1. **Fuel-air Ratio:** It is already mentioned that lean mixtures produce odours.
2. **Mode of engine operation:** The mode of operation of the engine affects the exhaust odour significantly. Maximum odour occurs when the engine is accelerated from idling.
3. **Engine type:** The odour intensity does not change with the type of engine—two-stroke or four-stroke engine.

It is also claimed by a few researchers that the intensity of odour is reduced by additive compounds.

Figure 11.25 *Effect of fuel air ratio on odour in diesel exhaust*

Unburned Hydrocarbons

The concentration of hydrocarbons in diesel exhaust varies from a few ppm to

a several thousand ppm, depending on the load on the engine and its speed. The hydrocarbons in diesel exhaust are composed of a mixture of many individual hydrocarbons in the fuel supplied to the engine as well as partly burned hydrocarbons produced during combustion process.

During the normal operation of the engine, the relatively cold wall quenches the fuel-mixture and inhibits the combustion, leaving a thick layer of unburned fuel air mixture over the entire surface of the combustion chamber. The thickness of this layer depends on the combustion pressure, temperature, mixture ratio, turbulence, and residual gases in the engine at the end of the

exhaust stroke. A greater surface to volume ratio of the combustion chamber leads to the formation of greater fraction of hydro-carbon from the quenched zone.

Carbon Monoxide

CO is formed when there is insufficient O_2 to completely oxidise the fuel during combustion of fuel. The amount of CO formed in a diesel engine is considerably lower than a petrol engine because of supply of continuous excess air to the engine. Theoretically, diesel engine should not emit CO at all as it always operates with excess air. However, CO is present in small quantities in diesel exhaust and this is, possibly, due to the fact that the fuel

injected during later part of injection does not find sufficient O_2 as a result of local depletion in certain parts of the combustion chamber.

The percentage of CO in exhaust varies from 0.1%–0.75%, which is easily acceptable level.

Oxides of Nitrogen

Among the gaseous pollutants emitted by the diesel engine, NO_x are the most significant. In this respect, the diesel engine is not very much behind the gasoline engine. NO_x being most hazardous, the limit is set to 350 ppm in many countries. In many diesel engines, NO_x varies from a few hundreds to 1000 ppm.

The mechanism of formation of NO_x is the same as discussed in the petrol engine. The conditions which create the highest local temperature (2000 K) and have sufficient O_2 give the highest NO_x concentration in diesel engine too.

The pre-combustion chamber diesel engines produce less NO_x than a direct injection engines because of low peak temperature. High fuel-air ratio (rich mixture), an additional fuel tends to cool the charge, and localised peak temperature falls and reduces the NO_x emission.

In addition, injection pattern, injection period, cetane number of the fuel, viscosity, and rate of burning also affect NO_x formation significantly. The effect

of load on NO_x emission rates for a four-stroke normally aspirated engine and four-stroke turbo-charged engines are shown in Fig. 11.26. The NO_x emission for turbocharged engine is considerably high compared with a normally aspirated engine at all load conditions.

Smog

The increase in the levels of SO_2, SO_3, NO_x, and suspended particulate matter creates havoc with the surrounding atmosphere. Out of these, SO_2 is the most potent as it causes bronchitis spasms. O_3 causes inflammations of the inner lining of air passages. A cocktail of these lethal substances narrows and inflames the air passages leads to

smelling in the lining of the throat and finally wheezing and difficulty in breathing.

Figure 11.26 *Effect of load on NOx emission for four-stroke normally and turbo-charged engines*

11.15.2 Control of Emission from Diesel Engine

The main pollutants from the diesel engine as mentioned earlier are HC, CO, NO_x smoke, odour, and SO_2. The methods of reducing HC, CO, and NO_x from petrol engine are already discussed in detail and the same methods are also used for reducing the pollutants from a diesel engine. Therefore, the methods used to reduce smoke and odour which are additional pollutants from diesel engine are discussed here.

Smoke and Control of Smoke

Formation of smoke is basically a process of conversion of molecules of hydrocarbon fuels into particles of soot. It should be noted that soot is not carbon but simply an agglomeration of very large polybenzenoid free radicals. It is also observed that soot formation during the early part of the actual combustion process is common to all diesel engines but is consumed during the latter part of combustion.

Pyrolysis of fuel molecules is thought to be responsible for soot formation. Fuel heated with insufficient O_2 will give carbonaceous deposits. It is believed that the 'heavy ends' of diesel fuel may pyrolyse to yield the type of smoke that is observed from a diesel engine. This is

believed to be the path of formation of polycyclic aromatic hydrocarbons (benzo-pyrene) found in soot.

Many theories have been put forward for the formation of smoke but the basic reactions leading to the formation of smoke are not fully known.

There is hardly any successful method to control the formation of soot except the engine has to run at lower load and maintain the engine at best possible condition.

Some methods suggested for the control of smoke are as follows.

1. **Smoke-suppressing additives:** It has been found that some barium compounds added in fuel reduce the temperature of combustion and avoid soot formation. It is further observed that if the soot is found, the barium compounds break them in very fine particles and reduce the smoke. However, berium

salts added in fuel form the deposit on engine parts and reduce the filtering capacity of the filter.

2. **Fumigation:** Fumigation is a method of injecting a small amount of fuel in the intake manifold. This helps pre-combustion reactions during compression stroke and reduces the chemical delay because the intermediate products such as peroxides and aldehydes react more rapidly with O_2 than hydrocarbons. Reducing the chemical delay curbs thermal cracking which is responsible for soot formation.

   The cracking may not even happen which is mainly responsible for soot formation when fumigation is used because it requires about 350 kJ/mole to break C-C bond and 425 kJ/mole to break C—H bond. The energy required may not be available due to easy oxidation during pre-combustion reaction.

3. **Catalytic convertors:** Catalytic convertors are not effective like in a petrol engine because of large soot formation which interferes in the oxidation of HC, CO, and NO_x. Therefore, these catalysts have a very small effect on engine smoke. Extensive research is being carried on to use catalysts for effective removal of soot as well as for removal of other emissions.

Odour Control

It is claimed by many manufacturers that odour additive compounds can reduce odour intensity. However, it is observed that by using additives, there is hardly any effect on odour formation

and is carried by exhaust gases.

The control of odours by using catalysts are under development and experiments have revealed that a few oxidation catalysts reduce odour intensity.

Unfortunately, the study of exhaust odour is hampered by lack of standard tests and standard units to measure the intensity of odour and the type of odour.

Several nations have been undertaking studies on suppressing odours by different methods.

Cleaning Up Diesel Emissions with Plasma and a Converter

While diesel engines are more economical, they produce NO_x during

combustion and put engine designers in a catch-22 situation. Presently, the permitted emission of NO_x in Europe is 0.7 g /km and it will be further reduced to 0.57 g /km. This limit is already in force in USA.

The major problem faced to reduce NO_x by catalytic converter which has proved successful in conventional engines cannot be applied to lean burn S.I. engines or diesel engines because these engines burn their fuel with high excess air (A:F = 30:1) and O_2 in the exhaust prevents the catalytic decomposition of NO_x.

Siemens and Partners have developed an efficient exhaust gas purification process for diesel engine ($SINO_x$). This system

comprises a catalytic converter, a control system, and a dosing device for urea. The urea undergoes hydrolysis into CO_2 and NH_3, which act as a reducing agent, transforming NO_x into environmentally compatible N_2 and water. However, in order to chemically reduce NO_x effectively in a catalytic converter, a minimum temperature of 200°C is essential—a condition that is usually met in trucks. In passenger cars, on other hand, the low efficiency of the process at temperatures below 200°C causes problems typical of urban driving and cold start phase.

11.15.3 NO_x–Emission Control

The concentration NO_x in the exhaust is closely related to the peak cycle

temperature. There are different methods by which peak cycle temperature can be reduced and NO_x emission can be controlled.

There are mainly three methods which are commonly used as follows: catalyst (which is already discussed), water injection (rarely used), and exhaust gas recirculation (EGR) method. EGR is commonly used to reduce NO_x. This method is used in petrol as well as diesel engines. In SI engines, about 10% recirculation reduces NO_x emission by 50%. Unfortunately, the consequently poorer combustion directly increases hydrocarbon emission and calls for mixture enrichment to restore combustion regularity which gives a

further indirect increase of both HC and CO.

Figure 11.27 shows the arrangement of an EGR system. A portion of the exhaust gases is recirculated to the cylinder intake charge. This reduces the quantity of O_2 available for combustion.

The exhaust gas for recirculation is taken as shown in Fig. 11.27 through an orifice and passed through the control valve for the regulation of the quantity of recirculation.

Figure 11.27 *EGR-system*

The effect of A:F ratio on NO_x emission takes EGR as a parameter as shown in Fig. 11.28. It can be seen that maximum emission of NO_x occurs during lean

mixture when gas recirculation is the least effective. On the other hand, for less emission of CO and HC, a lean mixture is preferred. About 15% recycling reduces NO_x by 80% but increases HC and CO by 50%–80%. These are conflicting requirements of this emission control system, and can be solved by adopting a package system to control all emissions.

Figure 11.28 *Effect of recycling of gas on NO_x concentration*

11.16 ❐ THREE-WAY CATALYTIC CONVERTER

A catalyst is a substance that accelerates a chemical reaction by lowering the energy needed for it to proceed. It is not consumed in the reaction. A three-way catalytic converter reduces the concentration of CO, HC, and NO_x in

the exhaust. A catalytic converter is usually a stainless steel container mounted along the exhaust pipe of the engine. There is a porous ceramic (Al_2O_3) structure inside the container through which the exhaust gas flows. The ceramic structure is a single honey-comb structure with many flow passages. The ceramic passages contain small embedded particles of catalytic material such as platinum, palladium, rhodium, and so on that promote oxidation or reduction reaction in the exhaust gas. Platinum and palladium promote the oxidation of CO and HC, whereas rhodium promotes the reaction of NO_x by reduction process.

The principle of working of a three-way

catalytic converter is shown in Fig. 11.29(a) and (b).

11.16.1 Function of a Catalyst in a Catalytic Converter

The catalyst controls the level of various exhaust pollutants from the engine by changing the chemical characteristics of the exhaust gases. Catalyst materials such as platinum or platinum, palladium, and rhodium are used in the converter. CO and HC oxidise the CO_2 and H_2O by palladium and platinum and NO_x is reduced by rhodium. An oxidation catalyst is placed downstream of the reduction catalyst.

Figure 11.29 *Three-way catalytic converter: (a) Catalytic converter package, (b) Three way catalytic converter*

where $z = x + 0.25\, y$

11.17 ❒ ENVIRONMENTAL PROBLEMS CREATED BY EXHAUST EMISSION FROM IC ENGINES

The emissions exhausted into the surroundings pollute the atmosphere and cause the following problems:

1. **Global warming:** The earth surrounding the atmosphere contains a 3 mm-thick layer of ozone (O_3) at 50 km from its surface in stratosphere. This layer of O_3 has the specific property to absorb ultraviolet (UV) rays emitted by the sun. If UV rays enter into the atmosphere and touch the earth, they will destroy all human, animal, and crop life. It has been observed that the O_3 layer is slowly getting destroyed due to the use of chlorofluorocarbons (CFCs) refrigerants used for refrigeration and air-conditioning purposes. The CFCs have verifying degree of ozone depletion potential (GWP) as well. The use of fully CFCs that are considered to have high ODP have been banned.
2. **Acid rain:** The diesel engine emissions contain oxides of sulphur, SO_x (SO_2 and SO_3) in varying amounts. These emissions are dissolved in water and give rise to sulphuric acid (H_2SO_4). They fall on the earth and are very harmful to human and plant life.
3. **Smog:** Smog means an increase in 'morbidity'. The increase in the levels of SO_2, SO_3, NO_x, and suspended particulate matter create huge problem in the surrounding atmosphere. SO_2 causes bronchitis spasms. SO_3 causes inflammations of the inner lining of air passages, swelling in the lining of the throat, and difficulty in breathing.
4. **Odours:** The members of the aldehyde family are considered

responsible for the pungent odours of diesel exhaust. Even 1 ppm aldehydes in exhaust can cause irritation to the nose and eyes. The intensity of the odour is reduced by additive compounds.

5. Respiratory and other health hazards

11.18 ❒ USE OF UNLEADED PETROL

It is desirable that a catalytic converter has an effective life time equal to half to the car life or at least 2 lakh km. Converters lose their efficiency with age due to thermal degradation and poisoning of the active catalyst material. Just a small amount of lead on a catalyst site reduces HC emission reduction by a factor of two of three. Use of leaded gasoline filled two times (full tank) would completely poison a converter and make it totally useless. Therefore, leaded gasoline cannot be used in engines equipped with catalytic

converters.

11.18.1 Use of Additives

To improve the combustion performance of fuels, some compounds called additives or dopes are used. The requirements of an additive are as follows:

1. It must be knock-resistant, surface ignition-resistant, or both.
2. It should be stable in storage and have no adverse effect on fuel stability.
3. It should be soluble in fuel under all conditions.
4. It should be in liquid phase at normal temperature and volatile to give rapid vapourisation in manifold.
5. It must not produce harmful deposits.
6. Its water solubility must be minimum to minimise handling losses.

The most commonly used additives are as follows:

1. Tetraethyl lead (TEL), ethylene dibromide (EDB), or ethylene dichloride (EDC) are added to TEL to avoid lead deposits.
2. Tetra-methyl lead (TML).

Summary for Quick Revision

1. **Performance parameters**
    1. Indicated power,
        1. p_{im} = indicated mean effective pressure (imep)
    2. Brake power,
    3. Frictional power, FP = IP − BP
    4. Mechanical efficiency,
    5. Specific output =
    6. Volumetric Efficiency,
    7. Brake specific fuel consumption,
    8. Indicated specific consumption,
    9. Brake thermal efficiency,
    10. Indicated thermal efficiency,
    11. Relative efficiency, =
    12. Fuel-air ratio,
    13. Relative fuel-air ratio,
    14. Equivalence ratio,
2. Willans line method is used to determine the FP of an engine. In this method, gross fuel consumption *v*'s BP at constant speed is plotted and the line is extrapolated back to zero fuel consumption. The point where the line cuts the BP axis gives the FP of the engine at that speed. This test is applicable to CI engines only.
3. **Morse test**

    This is a method to determine the IP of a cylinder in a multi-cylinder engine

    IP of n cylinders, $IP_n = BP_n + FP$

    IP of $(n - 1)$ cylinders, $IP_{n-1} = BP_{n-1} + FP$

    IP of nth cylinder, $IP_{n\text{th}} = BP_n - BP_{n-1}$

    Total IP of engine, $IP_n = \sum IP_{n\text{th}}$

4. **Air standard efficiency:**

5. **Heat balance sheet on minute basis:**

1. Heat supplied by fuel, $Q_s = \dot{m}_f \times CV$
2. Heat consumed in the system:
    1. Heat equivalent of BP, $Q_1 = 60 \times BP$ in kW, kJ/min
    2. Heat carried away by cooling water, $Q_2 = c_{pw} \times \dot{m}_w (T_{wo} - T_{wi})$, kJ/min
    3. Heat carried away by dry exhaust gases, $Q_3 = \dot{m}_g \times c_{pg} (T_{ge} - T_a)$, kJ/min

       where $\dot{m} = \dot{m}_a + \dot{m}_f$ = mass of exhaust gases, kg/min

       Steam in exhaust gases, $\dot{m}_s = 9 \times H_2 \times \dot{m}_f$ kg/min

       Mass of dry flue gases, $\dot{m}_g = \dot{m} - \dot{m}_s$

    4. Heat carried away by steam in exhaust gases,

       $Q_4 = \dot{m}_s [c_{pw} (100 - t_a) + h_{fg} + c_{ps} (t_{ge} - t_s)]$

    5. Heat unaccounted for, $Q_5 = Q_s - (Q_1 + Q_2 + Q_3 + Q_4)$

Multiple-choice Questions

1. Besides mean effective pressure, the data needed for determining the indicated power of an engine would include
    1. piston diameter, length of stroke, and calorific value of fuel
    2. piston diameter, specific fuel consumption, and calorific value of fuel
    3. piston diameter, length of stroke, and speed of rotation
    4. specific fuel consumption, speed of rotation, and

torque

2. For a typical automobile CI engine, for conditions of increasing engine speed, match List I with List II and select the correct answer using codes given below the lists:

Codes:

A B C

1. 1 2 3
2. 1 4 3
3. 2 3 4
4. 3 1 2

3. If the approximate average mean pressures during induction, compression, power and exhaust strokes of an internal combustion engine are, respectively, 15 kN/m^2 below atmosphere, 200 kN/m^2 above atmosphere, 1000 kN/m^2 above atmosphere and 20 kN/m^2 above atmosphere, then the resultant mean effective pressure, in kN/m^2, is
   1. 765
   2. 795
   3. 800
   4. 805

4. Match List I (Performance curves, labelled A, B, C and D, for a constant speed diesel engine) with List II (performance parameter) and select the correct answer using the codes given below the List:

Codes:

A B C D

1. 3 4 1 2
2. 3 4 2 1
3. 4 3 1 2
4. 4 3 2 2

5. Which one of following quantities is assumed constant for an internal combustion engine while estimating its friction power

by extrapolation through Willans line?

1. Brake thermal efficiency
2. Indicated thermal efficiency
3. Mechanical efficiency
4. Volumetric efficiency

6. A gas engine has a swept volume of 300 cc and clearance volume of 25 cc. Its volumetric efficiency is 0.88 and mechanical efficiency is 0.90. What is the volume of the mixture taken in per stroke?
    1. 248 cc
    2. 252 cc
    3. 264 cc
    4. 286 cc
7. The curve shown in the given Fig. 11.30 is characteristic of diesel engines.

    What does the Y-axis represent?

    1. Efficiency
    2. Specific fuel consumption
    3. Air-fuel ratio
    4. Total fuel consumption

Figure 11.30 *Characteristic of diesel engines*

8. The correct sequence of the decreasing order of brake thermal efficiency of the three given basic type of IC engines is
    1. four-stroke CI engine, four-stroke SI engine, two-stroke SI engine
    2. four-stroke SI engine, four-stroke CI engine, two-stroke SI engine
    3. four-stroke CI engine, two-stroke SI engine, four-stroke SI engine
    4. two-stroke SI engine, four-stroke SI engine, four-stroke CI engine
9. Keeping other parameters constant brake power diesel engine can be increased by
    1. decreasing the density of intake air
    2. increasing the temperature of intake air
    3. increasing the pressure of intake air
    4. decreasing the pressure of intake air
10. The method of determination of indicated power of multi-cylinder SI engine is by the use of
    1. Morse test

    2. Prony brake test
    3. motorist test
    4. heat balance test
11. In the context of performance evaluation of IC engine, match List I with List II and select the correct answer.

Codes:

A B C D

    1. 3 1 2 4
    2. 4 2 1 3
    3. 3 2 1 4
    4. 2 3 4 1
12. The presence of nitrogen in the products of combustion ensures that
    1. complete combustion of fuel takes place
    2. incomplete combustion of fuel occurs
    3. dry products of combustion are analysed
    4. air is used for combustion
13. A two-stroke engine has a speed of 750 rpm. A four-stroke engine having an identical cylinder size runs at 1500 rpm. The theoretical output of the two-stroke engine will
    1. be twice that of the four-stroke engine
    2. be half that of the four-stroke engine
    3. be the same as that of the four-stroke
    4. depend upon whether it is a CI or SI engine
14. For same power output and same compression ratio, as compared to two-stroke engines, four-stroke SI engines have
    1. higher fuel consumption
    2. lower thermal efficiency
    3. higher exhaust temperature
    4. higher thermal efficiency
15. Which one of the following plots correctly represents the variation of thermal efficiency (y-axis) with mixture strength (x-axis)?
    1.
    2.
    3.

4.

16. Match List I with the performance curves and select the correct answer using the codes given below the List:

Codes:

A B C D

1. 1 3 2 5
2. 1 3 2 4
3. 1 2 3 5
4. 2 1 4 3

17. Consider the following statements:
    1. Volumetric efficiency of diesel engines is higher than that of SI engines.
    2. When a SI engine is throttled; its mechanical efficiency decreases.
    3. Specific fuel consumption increases as the power capacity of the engine increases.
    4. In spite of higher compression ratios, the exhaust temperature in diesel engines is much lower than that in SI engines.

Of these statements,

1. I, II, III, and IV are correct
2. I, II, and III are correct
3. III and IV are correct
4. I, II, and IV are correct

18. In a variable speed SI engine, the maximum torque occurs at the maximum
    1. speed
    2. brake power
    3. indicated power
    4. volumetric efficiency
19. In a Morse test for a two-cylinder, two-stroke, spark ignition engine, the brake power was 9 kW, whereas the brake powers of individual cylinders with spark cut off were 4.25 kW and 3.75 kW, respectively. The mechanical efficiency of the engine is
    1. 90%
    2. 80%
    3. 45.5%

4. 52.5%

20. Match List I (performance Parameter Y) with List II (Curves labelled 1, 2, 3, 4, and 5 BHP vs. Y) regarding a CI engine run at constant speed and select the correct answer using the codes given below the lists:

Codes:

	A	B	C	D
1.	5	3	4	2
2.	1	3	4	2
3.	5	4	2	3
4.	1	4	2	3

21. Match List I with List II and select the correct answer using the codes given below the lists:

Codes:

	A	B	C	D
1.	4	1	2	5
2.	6	3	2	5
3.	6	1	5	2
4.	4	3	5	2

22. With respect to IC engine emissions, consider the following statements:
    1. Evaporative emissions have no carbon monoxide and oxides of nitrogen
    2. Blow-by emissions are essentially carbon monoxide and suspended particulate matter
    3. Exhaust emissions contain 100% of carbon monoxide, 100% of oxide of nitrogen, and around 50%–55% of hydrocarbons emitted by the engine
    4. There are no suspended particulate in the exhaust

Of these statements,

    1. I and IV are correct
    2. I and III are correct
    3. II and III are correct
    4. I, II, III, and IV are correct
23. A hydrocarbon fuel was burnt with air and the Orsat analysis of the dry products of combustion yielded the following data:

    The percentage (by volume) of CO_2 in the dry products was

    1. 2%
    2. 5%
    3. 11%
    4. 18%
24. The volumetric efficiency of a well-designed SI engine is in the range of
    1. 40%–50%
    2. 50%–60%
    3. 60%–70%
    4. 70%–90%
25. Variation of specific fuel consumption with fuel-air ratio for spark ignition engine is represented by which of the curves shown in Fig. 11.31?

    Figure 11.31

    1. Curve 1
    2. Curve 2
    3. Curve 3
    4. Curve 4
26. Exhaust emissions versus air-fuel ratio curves for a petrol engine are shown in Fig. 11.32.

    Figure 11.32

    The curve *C* represents

    1. hydrocarbon
    2. carbon dioxide
    3. carbon monoxide

4. oxides of nitrogen

27. If the performance of diesel engines of different sizes, cylinder dimensions, and power rating are to be compared, which of the following parameters can be used for such comparison?
    1. Swept volume
    2. Air fuel ratio
    3. Specific brake fuel consumption
    4. Volumetric efficiency
28. Consider the following statements for NO_x emissions from IC engines:
    1. Formation of NO_x depends upon combustion temperature
    2. Formation of NO_x depends upon type of coolant used
    3. Exhaust gas recirculation is an effective means for control of NO_x
    4. Activated Platinum is used for reduction of NO_x

    Which of the statements given above are correct?

    1. I and II
    2. I, II, and III
    3. II and IV
    4. I and III
29. Consider the following statements:

    Exhaust emission of carbon monoxide from spark ignition engine is

    1. Mainly fuel-air mixture strength dependent
    2. In the range of zero to 10%
    3. Measured with the help of an instrument working on the principle of non-dispersive infra-red analysis.
    4. Controlled by the use of a two way catalytic convertor

    Which of the statements given above are correct?

    1. I and IV
    2. II and III
    3. I and III
    4. I, II, III, and IV
30. An engine using octane-air mixture has N_2, O_2, CO_2, CO, and

H_2O as constituents in the exhaust gas. Which one of the following can be concluded?

1. Supply mixture is stoichiometric
2. Supply mixture has incomplete combustion
3. Supply mixture is rich
4. Supply mixture is lean

31. An engine produces 10 kW brake power while working with a brake thermal efficiency of 30%.If the calorific value of the fuel used is 40,000kJ/kg, then what is the fuel consumption?

    1. 1.5 kg/h
    2. 3.0 kg/h
    3. 0.3 kg/h
    4. .0 kg/h

32. A 40 kW engine has a mechanical efficiency of 80%. If the frictional power is assumed to be constant with load, what is the approximate value of the mechanical efficiency at 50% of the rated load?

    1. 45%
    2. 55%
    3. 65%
    4. 75%

33. Consider the following statements:

    1. Supercharging increases the power output and increases the volumetric efficiency
    2. Supercharging is more suitable for SI engines than CI engines
    3. The limit of supercharging for an SI engine is set by knock while that for a CI engine is set by thermal loading

    Which of the statements given above are correct?

    1. I and III
    2. I, II, and III
    3. II and III
    4. I and II

34. Which one of the following cannot be controlled by a three-way catalytic converter?

    1. HC emission
    2. CO emission
    3. NO_x emission
    4. SPM emission

35. The discharge of hydrocarbons from petrol automobile exhaust

is minimum when the vehicle is

1. idling
2. cruising
3. accelerating
4. decelerating

36. What is the purpose of employing supercharging for an engine?
    1. To provide forced cooling air
    2. To raise exhaust pressure
    3. To inject excess fuel for coping with higher load
    4. To supply an intake of air at a density greater than the density of the surrounding atmosphere
37. Consider the following statements:
    1. Supercharging increases the power output of an engine
    2. Supercharging increases the brake thermal efficiency considerably
    3. Supercharging helps scavenging of cylinders

Which of the statements given above are correct?

1. Only I and II
2. Only II and III
3. Only I and III
4. I, II, and III

Explanatory Notes

1. 3. (a) Resultant MEP = 1000 − 200 − (15 + 20) = 765 KN/m^2
2. 6. (c) Volumetric efficiency,

Volume of mixture per stroke = 300 × 0.88 = 264 cc

1. 19. (a) $IP_2 = 9 - 4.25 = 4.75$ kW

$IP_1 = 9 - 3.75 = 5.25$ kW

$IP = IP_1 + IP_2 = 4.75 + 5.25 = 10$ kW

Mechanical efficiency =

2. 23. (c) $CO_2 = 100 - 89 = 11$ cc, $O_2 = 89 - 84 = 5$ cc

$CO = 84 - 82 = 2$ cc

3. 32. (c)

At 50% rated load, η_{mech} =

Review Questions

1. Explain the method to measure the brake power of a small engine.
2. Describe the method to measure the heat lost in exhaust gases of an IC engine.
3. Explain the Morse test to measure the indicated power of a multi-cylinder engine.
4. Describe the method commonly used in laboratory to measure the air supplied to an IC engine.
5. Derive the formula used for finding the mass of air supplied to an engine using an orifice meter and tank.
6. Draw the following curves for a single-cylinder, four-stroke petrol, and diesel engines:
    1. BP *v*'s fuel consumption
    2. BP *v*'s SFC
    3. BP *v*'s brake thermal efficiency
7. Discuss the various performance parameters of IC engines.
8. What are the effects of load on the following?
    1. η_{vol}
    2. η_{mech}
    3. η_{bt}
    4. BSFC of SI engines
9. Define the following:
    1. BP
    2. IP
    3. FP
    4. BSFC
    5. η_r
10. Define the following:
    1. Equivalence ratio

2. ISFC
3. Specific weight
4. p_{im}

11. What are the categories of SI engine emissions?
12. Show the emissions from a four-wheeler having an SI engine on a neat sketch.
13. List the major exhaust emissions from a SI engine.
14. What is evaporative emission?
15. Name four methods for control of emissions in SI engines.
16. What is an after-burner? What is it used for?
17. What is a catalytic converter?
18. What is an evaporation emission control device?
19. Name the emissions from a CI engine.
20. Define smog and fumigation.
21. Explain SO_x and NO_x.
22. Name the catalyst materials used in a catalytic converter.
23. What are the environmental problems created by exhaust emission from IC engines?
24. What are additives used in IC engine fuels?
25. Name four alternative fuels for IC engines.
26. Define performance number.
27. Define HUCR.
28. List four desirable properties of IC engine fuels.
29. Define octane number and cetane number.
30. What do you understand by rating of fuels?
31. What are the main pollutants emitted by SI engine?
32. What are the various types of exhaust emissions from SI engine? Discuss briefly.
33. How exhaust emissions in SI engines can be controlled? Explain briefly.
34. What are the various types of exhaust emissions from CI engine? Explain briefly.
35. Describe the methods for control of emission from diesel engine.
36. Write a short note on No_x emission control.
37. What is the function of a catalytic converter? Explain the working of a three-way catalytic converter with the help of a neat sketch.
38. Write short notes on the following:
    1. Environmental problems created by exhaust emissions from IC engines.
    2. Use of Unleaded petrol.
    3. Use of additives in IC engine fuels.

39. Explain the various alternative fuels for IC engines.

Exercises

11.1 A single cylinder, four-stroke cycle oil engine works on diesel cycle. The following data is available from a test:

Area of indicator diagram = 3 cm^3, length of indicator diagram = 4 cm, spring constant = 10 bar/cm^2-cm, engine speed = 400 rpm, load on the brake = 380 N, spring reading = 50 N, diameter of brake drum = 1.2 m, fuel consumption = 2.8 kg/h, LCV = 42 MJ/kg, cylinder diameter = 16 cm, piston stroke = 20 cm.

Calculate (a) the friction power (b) the mechanical efficiency (c) the brake thermal efficiency, and (d) the brake

mean effective pressure.

[**Ans.** (a) 1.71 kW, (b) 83%, (c) 25%, (d) 6.15 bar]

11.2 A four-cylinder, four-stroke petrol engine 6 cm bore and 9 cm stroke was tested at constant speed. The fuel supply was fixed to 0.13 kg/min and spark plugs of 4-cylinders were successively short-circuited without change of speed.

The power measurements were as follows:

With all cylinder firing = 16.25 kW, with first cylinder cut-off = 11.55 kW,

With 2nd cylinder cut-off = 11.65 kW, with 3rd cylinder cut-off = 11.70 kW,

With 4th cylinder cut-off = 11.50 kW.

Determine (a) the indicated power of the engine, (b) the mechanical efficiency, (c) the indicated thermal efficiency if LCV of fuel used is 42 kJ/kg, and (d) the relative efficiency on IP basis assuming clearance volume 65 cm^3.

[**Ans.** 18.5 kW, 88%, 20.4%]

11.3 The following readings were taken during the test on a single cylinder, four-stroke IC engine:

Speed of engine = 240 rpm, orifice diameter of tank = 20 mm, pressure causing flow through the orifice = 10 cm of H_2O, ambient temperature = 30°C, barometer reading = 760 mm of Hg, gas consumption = 3.8 m^3/h at 30°C and 10 cm of H_2O above atmosphere.

Find (a) the air consumption per hour, and (b) the A:F ratio by volume and weight. Take $C_d = 0.7$ for air orifice, R = 287 J/kg.K for air and 280 J/kg.K for gas.

[**Ans.** (a) 0.544 m^3/min, (b) 85:1, 8:1]

11.4 The following data was obtained from a test on a single-cylinder, four-stroke oil engine.

Cylinder bore = 15 cm, stroke = 25 cm, area of indicator diagram = 540 mm^2, length of indicator diagram = 50 mm, indicator spring rating = 1.2 mm for a pressure of 8.8 N/cm^2, engine speed = 400 rpm, brake torque = 225 N-m, fuel consumption = 3 kg/h, calorific value of fuel = 44200 kJ/kg, cooling water flow rate = 4 kg/min, cooling water

temperature rise = 42°C, specific heat of water = 4.1868 kJ/kg.°C.

Calculate (a) the mechanical efficiency, (b) the brake thermal efficiency, (c) the specific fuel consumption, and (d) draw heat balance sheet.

[**Ans.** (a) 87.07%, (b) 25.6%, (c) 0.31 kg/kWh, (d) Q_s = 36.83 kW, Q_w = 11.72 kW, Q_{ge} = 15.68 kW, Q_b = 9.43 kW]

11.5 The following particulars were obtained in a trial for 1 h on a 4-stroke gas engine:

Speed = 16000 rpm, missed cycles = 600 rpm, net brake load = 1600 N, brake circumference = 4 m, IMEP = 8 bar, gas consumption = 2200 litres, CV of gas = 20 kJ/litre, d = 25 cm, L = 40 cm, r = 6.5.

Find (a) the indicated power and brake power (b) the brake specific fuel consumption, and (c) the brake thermal efficiency and relative efficiency.

[**Ans.** (a) 32.3 kW, 28.4 kW, (b) 773.4 litres/kWh, (c) 23.3%, 44%]

11.6 A 8-cylinder, 4-stroke petrol engine of 80 mm bore and 90 mm stroke has a compression ratio of 7. It is tested at 4000 rpm on a dynamometer which has 54 cm arm. During a 10 minute test, the dynamometer reads 400 N and the engine consumes 4.75 kg of fuel. Air is supplied at 1 bar and 27°C at the rate of 6.5 kg/min. CV of fuel = 44 MJ/kg.

Determine (a) the brake power (b) the brake mean effective pressure (c) the volumetric efficiency (d) the A:F ratio, and (e) the relative efficiency.

[**Ans.** (a) 90.8 kW, (b) 6 bar, (c) 77.32%, (d) 13.68:1, (e) 48%]

11.7 A six-cylinder gasoline engine operates on four stroke cycle. The bore of each cylinder is 80 mm and the stroke is 100 mm. At speed of 4000 rpm the fuel consumption is 20 kg/h and the torque developed is 150 Nm. Calculate (*i*) the brake power (*ii*) the brake mean effective pressure (*iii*) brake thermal efficiency if calorific value of the fuel is 43000 kJ/kg.

[**Ans.** (a) 62.83 kW, (b) 6.25 bar, (c) 26.3%]

11.8 An eight-cylinder, four-stroke automobile engine develops 76.25 kW brake power when tested at 4000 rpm. The compression ratio of the engine is 7. The testing is carried out for 10 minutes and fuel consumption was 4.5 kg. CV of

fuel used = 45000 kJ/kg. The air passing through the carburettor has 20°C temperature and 1.03 bar pressure, Air measured was 5.45 kg/min. The diameter and stroke are both equal to 8.5 cm. Determine (a) the brake mean effective pressure, (b) the brake specific fuel consumption, (c) the brake thermal efficiency, and (d) the A:F ratio.

[**Ans.** (a) 5.93 bar, (b) 0.354 kg/kWh, (c) 57.6%, (d) 12.25:1]

11.9 A six-cylinder, four-stroke petrol engine develops 40 kW when running at 3000 rpm. The volumetric efficiency at NTP is 85% and the indicated thermal efficiency is 25%. The A:F ratio of the mixture supplied is 15:1. The calorific value of fuel used is 41000 kJ/kg. Assume the diameter to be equal to

stroke of the engine, calculate the bore and stroke.

[**Ans.** 76 mm, 76 mm]

11.10 A single cylinder, four-stroke cycle oil engine works on diesel cycle. The following readings were taken when the engine was running at full load:

Area of indicator diagram = 3 cm, length of diagram = 4 cm,

spring constant = 10 bar/cm^2. cm, speed of engine = 400 rpm,

Load on the brake = 380 N, Spring reading = 50 N, diameter of brake drum = 1.2 m, fuel consumption = 2.8 kg/h, calorific value of fuel = 42000 kJ/kg,

cylinder diameter = 16 cm, stroke = 20 cm. Calculate (a) the friction power of the engine, (b) the mechanical efficiency, (c) the brake thermal efficiency, and (d) the brake mean effective pressure.

[**Ans.** (a) 1.71 kW, (b) 83%, (c) 25%, (d) 6.15 bar]

11.11 A four stroke petrol engine 80 mm bore 100 mm stroke is tested at full throttle at constant speed. The fuel supply is fixed at 0.08 kg/min and the plugs of the four cylinders are successively short circuited without change of speed, brake torque being correspondingly adjusted. The brake power measurements are the following:

With all cylinder firing = 12.5 kW

With cylinder No. 1 cut-off = 9 kW

With cylinder No. 2 cut-off = 9.15 kW

With cylinder No. 3 cut-off = 9.2 kW

With cylinder No. 4 cut-off = 9.1 kW

Determine the indicated power of the engine under these conditions. Also determine the indicated thermal efficiency. Calorific value of the fuel is 44100 kJ/kg. Compare this efficiency with the air standard value, clearance volume of one cylinder is 70×10^3 mm^3.

[**Ans.** 13.55 kW, 23.04%, 40.52%]

11.12 A four-cylinder, four-stroke petrol engine, 60 mm bore and 90 mm stroke was tested at constant speed. The fuel

supply was fixed at 0.13 kg/min and plugs of 4-cylinders were successively short-circuited without change of speed. The power measurements were as follows:

With all cylinders working = 16.25 kW

With 1st-cylinder cut-off = 11.55 kW (BP)

With 2nd-cylinder cut-off = 11.65 kW (BP)

With 3rd-cylinder cut-off = 11.70 kW (BP)

With 4th-cylinder cut-off = 11.50 kW (BP)

Find the IP of engine, the mechanical efficiency, the indicated thermal efficiency if CV of fuel used is 42,000 kJ/kg, and the relative efficiency on IP basis assuming clearance volume 65 m^3.

[**Ans.** 1850 kW, 88%, 20.4%, 42%]

11.13 The air flow to a four cylinder, four-stroke petrol engine is measured by means of a 7.5 cm sharp-edged orifice, $C_d = 0.6$. During a test on the engine, following data was recorded.

Bore = 12 cm, stroke = 14 cm, engine speed = 2200 rpm, brake power = 35 kW, fuel consumption = 10 kg/h, LCV = 42000 kJ/kg, pressure drop across the orifice = 4 cm of water. Atmospheric temperature = 20°C, and atmospheric pressure = 1.01325 bar.

Calculate (a) the thermal efficiency on BP basis, (b) the brake mean effective pressure, and (c) the volumetric efficiency based on free air conditions.

[**Ans.** (a) 30%, (b) 1.507 bar, (c) 70.2%]

11.14 The following readings were taken during test on a single cylinder, four-stroke gas engine: speed of engine = 240 rpm, diameter of orifice of air tank = 20 mm, pressure causing the air flow through orifice = 10 cm of water, ambient temperature = 30°C, barometer reading = 76 cm of Hg, gas consumption = 3.8 m^3/h measured at 30°C and 10 cm of water above atmosphere.

Find (a) air consumption per hour and (b) the A:F ratio by weight and by

volume. Take $C_{da} = 0.7$, $R = 287$ J/kg. K for air, and $R = 280$ J/kg.K for gas

[**Ans.** (a) 0.544 m^3/min, (b) 8.5:1, 8.1]

11.15 The following particulars were obtained in a trial on a 4-stroke gas engine:

Duration of trial = 1 h, revolutions = 16000, missed cycles = 600, net brake load = 1600 N, brake circumference = 4 m, MEP = 8 bar, gas consumption = 2200 litres, CV of gas = 20 kJ/litre, D = 25 cm, L = 40 cm compression ratio = 6.5. Calculate the indicated power, the brake power, the brake specific fuel consumption, the brake thermal efficiency, and the relative efficiency.

[**Ans.** 32.3 kW, 28.4 kW, 773.4 litres/kWh, 23.3%, 44%]

11.16 The following data is given for a four-stroke, four-cylinder, diesel engine:

Cylinder diameter = 35 cm, piston stroke = 40 cm, engine speed = 315 rpm, IMEP = 7 bar, BP of engine = 260 kW, fuel consumption = 80 kg/h, CV of fuel used = 43000 kJ/kg, hydrogen content in fuel = 13% and remaining is carbon, air consumption = 30 kg/min, cooling water circulated = 90 kg/min, rise in temperature of cooling water = 38°C, piston cooling oil used = 45 kg/min, rise in temperature of cooling oil = 23°C, c_p for cooling oil = 2.2 kJ/kg.K, exhaust gas temperature = 322°C, c_p for exhaust gases = 1.1 kJ/kg.K, ambient temperature = 22°C, c_p of superheated steam = 2 kJ/kg.K, latent heat of steam

= 2520 kJ/kg.

Calculate (a) the mechanical and indicated thermal efficiency, (b) find bsfc when load on the engine is 50% of full load assuming same indicated thermal efficiency, and (c) draw up heat balance sheet on minute and percentage basis.

[**Ans**. (a) 92%, 29.6%, (b) 0.308 kg/kWh, (c) Q_s = 57333 kJ/min, Q_b = 15600 kJ/min, Q_w = 14364 kJ/min, Q_{oil} = 2277 kJ/min, Q_{ge} = 9858 kJ/min, Q_{steam} = 4540 kJ/min, Q_{unacc} = 10694 kJ/min]

ANSWERS TO MULTIPLE-CHOICE QUESTIONS

1. c
2. d
3. a
4. a
5. b
6. c
7. d
8. a
9. c
10. a
11. c
12. d
13. c
14. d
15. c

16. a
17. a
18. c
19. a
20. a
21. a
22. c
23. c
24. d
25. c
26. d
27. c
28. c
29. c
30. b
31. b
32. c
33. b
34. a
35. c
36. d
37. c

Chapter 12

Reciprocating Air Compressors

12.1 ❐ INTRODUCTION

Compressors are mechanical devices used for increasing the pressure of a gas. Compressors used for producing high pressure air are called air compressors. Air is drawn from the atmosphere by suction process, which is then compressed to the required pressure and delivered to the receiver. Air compressors may be classified as shown in Fig. 12.1.

Figure 12.1 *Classification of compressors*

If compression is done in a conventional

cylinder with a closely fitted piston making reciprocating motion, then the compressor is called a *reciprocating compressor*. External work must be supplied from a prime mover to the compressor to achieve the required compression. The general arrangement of the compressor and prime mover used to run the compressor is shown in Fig. 12.2.

Figure 12.2 *Energy flow in an air-compressor*

12.2 ❐ USES OF COMPRESSED AIR IN INDUSTRY

The several uses of compressed air are as follows:

1. To drive a compressed air engine
2. For producing an air blast for a workshop
3. For spraying the fuel (atomizing) into a boiler furnace
4. For operating pneumatic drills and tools
5. For operating pneumatic brakes for locomotives and rolling stock
6. Pneumatic conveying and for pumping of water by compressed air

7. For working compressed air engines especially in mines
8. In gas turbine power plants and air-conditioning plants

12.3 ❒ WORKING PRINCIPLE OF SINGLE-STAGE RECIPROCATING COMPRESSOR

The schematic arrangement of a single-stage reciprocating compressor is shown in Fig. 12.3. It consists of a cylinder with cooling jacket, piston, connecting rod, crank, suction valve, and delivery valve. The piston will be making to-and-fro motions through the crankshaft arrangement and generally run by an electric motor, diesel engine, petrol engine, or steam engine. During the outward motion of the piston, the pressure inside the cylinder falls below the atmospheric pressure and the suction valve is opened due to the pressure difference. The atmospheric air is then drawn into the cylinder until the piston

reaches the bottom dead centre position.

Figure 12.3 *Single-stage reciprocating air-compressor*

As the piston starts move inwards, the suction valve gets closed and the pressure starts increasing until the pressure inside the cylinder is more than the pressure of the delivery side. Then the delivery valve opens and high pressure air is delivered to the receiver till the piston reaches the top dead centre. At the end of the delivery stroke, the small volume of high pressure air left in the clearance volume expands as the piston starts moving outwards. Hence, the cycle is repeated.

12.4 ❐ TERMINOLOGY

1. **Single-acting compressor:** If the air admission from the atmosphere is on only one side of the cylinder.
2. **Double-acting compressor:** When, the air from the atmosphere is drawn on both sides of the piston.

3. **Single-stage compressor:** If the total compression is done fully in one cylinder.
4. **Multi-stage compressor:** If the compression is carried out in more than one cylinder, and every cylinder carries out a part of the compression.
5. **Free Air Delivered (FAD):** It is the actual volume delivered at the stated pressure, reduced to intake pressure and temperature, and expressed in cubic metre/minute (m3/min). FAD is used to measure the capacity of a compressor.
6. **Displacement or Swept Volume of the Compressor:** The volume displaced by piston movement between two dead centers is called displacement or swept volume. For a single acting compressor, swept volume,

   where d = diameter of piston and L = length of stroke

12.5 ❒ TYPES OF COMPRESSION

The theoretical $p-V$ diagram for single acting compressor is shown in Fig.12.4. The line 4–1 represents the suction stroke. The air is then compressed adiabatically as shown by the curve 1–2 in Fig. 12.4. It is then forced out of the cylinder at constant pressure p_2, as shown by the line 2–3. The work done is represented by the area 1–2–3–4–1.

Figure 12.4 *Theoretical p-V diagram for single acting compressor*

If the air had been compressed isothermally, as represented by the curve 1–2′, then the work done on the air would be the area 1–2′–3–4–1, which is considerably less than that due to adiabatic compression. However, it is not possible in practice to compress the air isothermally because in that case, the compressor would need to run extremely slow. In practice, the compressors are driven at fairly high speeds in order to compress as much air as possible in a given time. Hence, the compression of air will approximate to an adiabatic. The work saved by compressing isothermally is shown by the shaded area 1–2–2′–1.

12.5.1 Methods for Approximating Compression Process to Isothermal

The following practical methods are used to achieve nearly isothermal compression:

1. **Cold water spray:** In this method, cold water is sprayed onto the cylinder during compression, thus reducing the temperature of the air. Without the cold water spray, the compression would have been adiabatic or pV^{γ} = constant. This effect is shown in Fig. 12.5, where now the compression would be between adiabatic and isothermal or pV^{n} = const., where $1 < n < 1.4$. The work saved is represented by 1-2-2″-1.

Figure 12.5 *Effect of type of compression on work done.*

Figure 12.6 *Multi-stage compression*

2. **Water jacketing:** The water is circulated around the cylinder through the water jacket to cool the air during compression. This method is commonly used for all types of reciprocating compressors.
3. **Intercooling by using multi-stage compression:** In a multi-stage compressor, the air is compressed in several stages. In principle, it is equivalent to number of compressors in series where the air passes from one cylinder to the next and the pressure increases in each cylinder. The $p - V$ diagram for a four-stage compressor is shown in Fig. 12.6. The dotted line 1–4′ is the isothermal. In the first stage, the air is compressed adiabatically to 2′ and cooled at constant pressure to 2′ in the intercooler, possibly to the initial temperature. For complete intercooling, the point 2′ is on the isothermal line 1–4′. The air is then drawn into the second cylinder for the second stage of compression and the process is repeated for the subsequent cylinders. Line 1–*a* represents the adiabatic compression in a single cylinder. The compressor work saved by intercooling is represented by the area 2–*a*–4–3′–3–2′–2.
4. **External fins:** Small capacity compressors are provided with fins to increase the heat transfer from the surface of the cylinder.

12.6 ❐ SINGLE-STAGE COMPRESSION

12.6.1 Required Work

Without Clearance

Consider the theoretical p–V diagram for a single-stage air compressor as shown in Fig. 12.7. The work done on air per cycle is represented by the area 1–2–3–4–1.

Figure 12.7 *p-V diagram without clearance*

W = area ($a - b - 2 - 3 - a$) + area ($b - c - 1 - 2 - b$) – area ($a - c - 1 - 4 - a$)

where m = mass of air delivered per cycle.

Work done per kg of air

If the compression had followed the law, $pV^n = c$, then

Power required in driving the compressor

where N = number of complete cycles per minute

= rpm, if single acting

= number of strokes per minute, if double acting.

With Clearance Volume

The p-V diagram for a single-stage and single-acting air compressor is shown in Fig. 12.8 with clearance volume.

Let V_c = clearance volume

V_s = swept volume = $V_1 - V_c$

V_a = actual volume = $V_1 - V_4$

Let the compression and expansion processes follow the same low pV^n = const.

Work done per cycle,

$$W = \text{area } (1\text{–}2\text{–}3\text{–}4\text{–}1)$$
$$= \text{area } (1\text{–}2\text{–}5\text{–}6\text{–}1) - \text{area } (3\text{–}4\text{–}6\text{–}5\text{–}3)$$

Now, $p_3 = p_2$ and $p_4 = p_1$

Figure 12.8 *p-V diagram with clearance*

where m_1 = actual mass of air delivered per cycle.

12.6.2 Volumetric Efficiency

The *volumetric efficiency* of a

reciprocating compressor is defined as the ratio of the actual free air delivered to the swept volume of the compressor. The free air delivered is (V_1–V_4), whereas the swept volume is (V_1–V_c). Thus,

Let clearance ratio

When referred to ambient conditions,

Factors Affecting Volumetric Efficiency

The volumetric efficiency of a compressor can be lowered by any of the following conditions:

1. Very high speed
2. Leakage through piston seals
3. Too large a clearance volume
4. Obstruction at inlet valves
5. Overheating of air by contact with cylinder walls

6. Inertia effect of air

Figure 12.9 shows the variation in volumetric efficiency with clearance ratio (c), and the pressure ratio (p_2/p_1), and polytropic index of compression (n) (by changing one factor keeping the other two factors constant). The volumetric efficiency decreases with increase in both the clearance ratio (c), and the pressure ratio (p_2/p_1), whereas it increases with increase in the polytropic index of compression (n).

Figure 12.9 *Variation of volumetric efficiency with clearance ratio, pressure ratio and polytropic index of compression*

12.6.3 Isothermal Efficiency

The p-V and T- s diagrams for isothermal and polytropic compression of air in the compressor, respectively,

are shown in Fig. 12.10.

Polytropic work done,

Figure 12.10 *Isothermal and polytropic compression in a compressor*

Isothermal work done

However, $p_1v_1 = p_2v_{2'}$

Work saved, $\Delta W = W_p - W_i$

The isothermal efficiency (η_i)is a measure of the degree to which isothermal compression has been achieved. It is defined as the ratio of isothermal work to that of actual indicated work and is given by

12.6.4 Adiabatic Efficiency

It is defined as the ratio of the work done on the compressor with reversible adiabatic compression to the work done with irreversible polytropic compression.

12.6.5 Calculation of Main Dimensions

The actual volume of air drawn in by the compressor per stroke,

Capacity of a single-acting compressor,

Also,

where d = piston diameter

L = length of stroke

Therefore, L and d can be determined.

12.7 ❒ MULTI-STAGE COMPRESSION

Single-stage compression suffers from many disadvantages such as (i) handling of very high pressure range in one cylinder resulting in leakage past the piston, (ii) ineffective cooling of the gas, and (iii) necessitating robust construction of the cylinder to withstand the high delivery pressure.

The volumetric efficiency of a single-stage compressor with fixed clearance decreases with an increase in pressure ratio and thus reduces the capacity. Thus the necessity of of multi-stage compression with intercooling between stages are listed below:

1. The air can be cooled perfectly at pressures intermediate between the suction and delivery pressures resulting in less power required as compared to a single-stage compressor for the same pressure limits and quantity of free air delivered.
2. The mechanical balance of the machine is better due to phasing of the operations.
3. The pressure range and hence the temperature range in each stage can be kept within desirable limits. This results in:
    1. Less loss of air due to leakage past the piston.
    2. More perfect lubrication due to lower temperatures.
    3. Better volumetric efficiency.
    4. The lighter construction of the machine that reduces the cost.

12.7.1 Two-stage Compressor

The schematic arrangement of two-stage compressor with intercooler and $p - V$ diagram are shown in Fig. 12.11. The air is first taken into the low pressure (L.P.) cylinder at pressure p_1. After compression to intermediate pressure p_2, the air at condition 2 is passed through the intercooler and leaves it at point 3, where its temperature is reduced from T_2 to T_3. The air may be cooled to point 3′ such that $T_3' = T_1$. Finally, the air is compressed in high pressure (H.P.)

cylinder from condition 3 to 4 and is discharged to the receiver. Area 2–3–4–6–2 represents the work saved due to intercooling.

Figure 12.12 shows the $p - V$ diagram for two-stage compression with perfect intercooling. Process 1–2 represents the polytropic compression in the L.P. cylinder with pV^{n_1} = constant. Process 2–3 represents intercooling from T_2 to $T_3 = T_1$. Process 3–4 represents the polytropic compression in the H.P. cylinder with pV^{n_2} = constant. Curve 1–3–4′ represents the isothermal compression.

Total work done on the air,

Let and

Figure 12.11 *Two stage compression with intercooling: (a) Schematic arrangement, (b) p-V diagram*

Figure 12.12 *Two stage compression with perfect intercooling*

For p_1 and p_3 to be constant, intermediate pressure p_2 must be determined for minimum work.

Thus,

For $n_1 = n_2 = n$, and $a = b$.

For perfect intercooling, $T_3 = T_1$. Thus,

In general, with i number of stages, we have

The minimum indicated powers (I.P.),

with imperfect intercooling is:

where m = mass of air delivered in kg per second

The number of stage is decided by the delivery pressure for a given inlet pressure (1 bar) as follows:

1. For delivery pressure upto 5 bar: Single stage compressor
2. For delivery pressure between 5 to 35 bar: Two stage compressor
3. For delivery pressure between 35 to 85 bar: Three stage compressor
4. For delivery pressure more than 85 bar: Four stage compressor

12.7.2 Heat Rejected to the Intercooler

Let m = mass of air in the cylinder

Then $p_1 V_1 = m\mathrm{RT}_1$

or

From the compression 1–2, we have

From the constant pressure line 2–3, we have

12.7.3 Cylinder Dimensions

For steady-state flow, the mass of air passing through each cylinder per stroke must be same.

Let V_a = actual volume of air per stroke taken during suction

ρ = density of air

Then, $V_{a_1} \rho_1 = V_{a_2} \rho_2 = V_{a_3} \rho_3 = \ldots =$ const.

For perfect intercooling, $T_1 = T_2 = T_3 = \ldots$

$\therefore V_{a_1} p_1 = V_{a_3} p_2 = V_{a_3} p_3 = \ldots$

$$V_{s_1} \eta_{v_1} p_1 = V_{s_2} \eta_{v_2} p_2 = V_{s_3} \eta_{v_3} p_3 = \ldots$$

where η_v = volumetric efficiency of a cylinder

V_s = stroke volume of the cylinder

12.7.4 Intercooler and Aftercooler

Intercooler

An intercooler is a simple heat exchanger in which heat is removed from the air after it has been compressed and its temperature has risen as a result of compression. The intercooler commonly used is of the counter flow

type as shown in Fig. 12.13 because it gives high effectiveness.

Effectiveness of intercooler,

A simple section showing the principles of construction of an intercooler is shown in Fig. 12.14. The coolant, which may be water or any other fluid, passes through the tubes secured between two end plates and the air circulates over the tubes through a system of baffles. Two passes are used for water flow and air is made to flow partly parallel and partly cross with the help of baffles. This type of intercooler gives better effectiveness than the ordinary counter-flow type. The purpose of the intercooler is to reduce the work done on the air.

Figure 12.13 *Counter flow intercooler*

Figure 12.14 *Principles of construction of inter-cooler*

Aftercooler

An aftercooler is used to cool the air coming out from the compressor before it enters the receiver. The air coming out from the compressor at pressure p_3 is sufficiently hot at temperature T_4 (Fig. 12.15). If this air is cooled in the aftercooler, then the pressure will remain p_3, but the temperature will fall from T_4 to T_4''. Therefore, the volume of air leaving the aftercooler will be given by,

However, $p_4'' = p_4$.

As $T_4'' < T_4$, $v_4'' < v_4$

Therefore, the purpose of the aftercooler is to reduce the size of the receiver. The position of the intercooler and aftercooler is shown in Fig. 12.15.

Figure 12.15 *Position of inter-cooler and after-cooler*

12.8 ❐ INDICATED POWER OF A COMPRESSOR

where A = area of the cylinder, m^2

L = length of stroke, m

N = *rpm* of compressor crank

p_m = mean effective pressure of air, P_a (N/m^2)

Theoretically, *m.e.p.* for a single-acting, single-stage compressor is,

where η_v = volumetric efficiency of

compressor

Using indicator card, the *m.e.p.* is:

12.9 ❒ AIR MOTORS

The working of an air motor is similar to that of air compressor. High pressure air is admitted to the motor cylinder through a mechanically operated inlet valve and drives the piston in the forward direction. After a part of the stroke of the piston has been performed, the air supply is cut-off and the stroke is completed after decreasing pressure as the air expands in the cylinder. After the expansion stroke is completed, the air is allowed to escape into the atmosphere through a mechanically operated discharge valve. The return stroke is

performed by compressed air acting on the other side of the piston in double-acting cylinder.

Figure 12.16 shows the $p - V$ diagram for the air motor with clearance. Process 4–1 is the intake with the high pressure p_2 at cut-off point 1. From 1 to 2, the air expands from higher pressure p_2 at 1 to atmospheric pressure p_1 at 2, according to the polytropic law, $pV^n = c$. The process 2–3 represents the exhaust.

Figure 12.16 *p−V diagram for an air motor*

Work obtained neglecting clearance,

Work done considering clearance,

where m_1 = actual mass of air delivered

per cycle.

12.10 ❐ INDICATOR DIAGRAM

The theoretical of $p - V$ diagram for the single-stage reciprocating compressor is represented by 1–2–3–4–1 in Fig. 12.17. The actual indicator diagram is 1–2′–3–4′–1. The difference between the actual and theoretical indicator diagrams is due to the intake and discharge losses. The intake losses include the friction losses in pipe, friction loss in inlet valve, and valve inertia loss. Theoretically, the inlet valve should open at 4, but actually, it opens little afterwards at 4′ due to the inertia of the valve, and the pressure inside the cylinder falls below the atmospheric pressure. The oscillating part of the curve indicates

valve flutter due to vibration of spring loaded valve. On the delivery side, the discharge valve should open at 2, but actually, it opens little afterwards at 2′ due to the inertia of the spring loaded discharge valve. The effect of these losses is to increase the work required by the compressor.

Figure 12.17 *Actual p−V diagram for single stage compressor*

Figure 12.18 *Actual p−V diagram for two-stage compressor*

The actual indicator diagram for two-stage compressor with intercooler is shown in Fig. 12.18.

12.11 ❐ HEAT REJECTED

If the air is cooled to the initial temperature, then there is no change in internal energy per kg mass of air, and all the work done is rejected to the

cooling medium party during the compression process and the remaining after compression at constant pressure.

Now, $q_{1-2} = du + w_{1-2}$

However, $du = 0$

For a single-stage compressor, the heat rejected is given by work done - $\int vdp$.

= Heat rejected to the coolant in the intercooler + Heat rejected to the coolant during compression process

12.12 ❒ CONTROL OF COMPRESSOR

In order to balance the demand and supply of air, it is necessary to incorporate devices for the compressor control. The common methods of control are as follows:

1. Throttle control
2. Clearance control

3. Blowing air to waste

1. **Throttle control:** When the demand is less, there is build-up of pressure in the receiver, and the high pressure air from the receiver is led to piston and cylinder. The movement of piston is resisted by a spring. However, with excessive pressure, the piston depresses the spring, thus closing partly the suction valve. Therefore, during the suction stroke, the air intake is partly throttled. The reverse action takes place when the pressure in the receiver drops due to increase in demand.
2. **Clearance control:** In this method, clearance pockets are provided for increasing or decreasing clearance. Therefore, the volumetric efficiency is reduced in proper proportion to control output.
3. **Blowing air to waste:** In case of excessive pressure built-up in the receiver due to decrease in demand, a by-pass valve from the high pressure cylinder delivers air directly to atmosphere. When the pressure in the receiver drops, the relay piston closes the valve.

Example 12.1

A single-stage reciprocating air compressor is required to compress 72 m^3 of air per minute from 15°C and 1.0 bar to 8 bar pressure. Find the temperature at the end of compression, work done, power, and heat rejected during each of the

following processes: (a) isothermal, (b) adiabatic, and (c) polytropic compression following the law $pV^{1.25}$ = constant.

Solution

1. **Isothermal compression:**
2. **Adiabatic compression:**

   Heat rejected = 0

3. **Polytropic compression:**

Note: Example 9.1 illustrates that isothermal compression requires the minimum compression work, while isentropic compression requires the maximum for the same suction and delivery pressure.

Example 12.2

A double-acting reciprocating compressor with a piston displacement of 0.05 m^3 per stroke operates at 500 rpm. The clearance is 5 per cent and it receives air at 100 kPa, discharges it at 600 kPa. The compression is polytropic, $pV^{1.35} = c$. Determine the power required and the air discharged in m^3/s.

Solution

The *p–V* diagram for the compressor is shown in Fig. 12.19.

$$V_a = V_c + V_s - V_4$$
$$V_c = 0.05 \times 0.05 = 0.0025 \text{ m}^3$$

Figure 12.19 *p–V diagram*

Example 12.3

A double-acting reciprocating compressor with complete intercooling delivers air to the main at a pressure of 30 atm, the suction condition being 1 atm and 27°C. If both cylinders have the same stroke, then find the ratio of the diameters of the cylinders for the efficiency of compression to be a maximum. Assume the index of compression to be 1.3.

Solution

Volume of L.P. cylinder = V_1

Volume of H.P. cylinder = V_3

$L_1 = L_3$

For maximum efficiency,

From constant pressure process 2–3: (see Fig. 12.12)

Example 12.4

A double-acting reciprocating compressor with perfect intercooling takes in air at 1 bar and 27°C. The law of compression in both the stages is $pV^{1.3}$ = constant. The compressed air is delivered at 9 bar from the H.P. cylinder to an air receiver. Calculate per kg of air (a)

the minimum work done, (b) the heat rejected to the intercooler, and (c) the minimum work done in a three-stage compressor working under the same conditions. Take c_p = 1.005 kJ/kg.K.

Solution

$$T_1 = 273 + 27 = 300 \text{ k}$$

For 1 kg of air, $p_1 v_1 = RT_1$,

or $v_1 = 0.861 \text{ m}^3/\text{kg}$

1. The minimum work required in two-stage compressor with perfect intercooling is,

   The intermediate pressure is found to be

2. Heat rejected to intercooler,

$$q = c_p (T_2 - T_3) = \text{cp } (T2 - T_1) \; [\because T_2 = T_3]$$
$$= 1.005(386.6 - 300) = 87 \text{ kJ/kg of air}$$

3. The least work done in case of three-stage air compressor working between the same pressure limits is given by,

Example 12.5

A single-stage, single-acting reciprocating compressor delivers 15 m^3/min of free air from 1 bar to 8 bar at 300 rpm. The clearance volume is 6% of the stroke volume, and compression and expansion follow the law $pV^{1.3}$ = constant. Calculate the diameter and stroke of the compressor. Take $L = 1.5\ D$. The temperature and pressure of air at suction are the same as atmospheric air.

Solution

Clearance ratio, $c = 0.06$

Volumetric efficiency,

$$D = 0.3817 \text{ m or } 38.17 \text{ mm}$$
$$L = 1.5 \times 0.3817 = 0.5726 \text{ m or } 57.26 \text{ mm}$$

Example 12.6

A two-cylinder, single-acting reciprocating air compressor is to deliver 15 kg of air per minute at 6.5 bar from suction conditions 1 bar and 13°C. Clearance may be taken as 4 per cent of stroke volume and the index for both compression and re-expansion as 1.3. The compressor is directly coupled to a four-cylinder, four-stroke petrol engine that runs at 1800 rpm with b.m.e.p. of 6 bar. Assuming a stroke–bore ratio of 1.1 for both

engine and compressor and a mechanical efficiency of 85% for the compressor, calculate the required cylinder dimensions. Take $R = 287$ J/kgK.

Solution

The $p-V$ diagram for the compressor is shown in Fig. 12.20.

Amount of air delivered per cylinder

Now, $p_1(V_1 - V_4) = mRT_1$

From the expansion curve, we have

$$V_3 = 0.04\ V_s$$
$$V_4 = 0.1688\ V_s$$

Now, $V_1 = V_3 + V_s = 0.04\ V_s + V_s = 1.04\ V_s$

$$V_1 - V_4 = (1.04 - 0.1688)\ V_s = 0.8712\ V_s = 3.42 \times 10^{-3}$$
$$V_s = 3.9256 \times 10^{-3}\ \text{m}^3$$
$$D = 0.1656 \text{ m or } 165.6 \text{ mm}$$
$$L = 1.1\ D = 0.1822 \text{ m or } 182.2 \text{ mm}$$

Figure 12.20 *p-V diagram*

Example 12.7

A multistage reciprocating compressor has to be designed to supply air at 135 bar, while atmospheric condition is 1.03 bar and 15°C. The value of compression index may be assumed as 1.35. Due to practical reasons, the intercoolers are not able to cool the air below 45°C, while the maximum temperature allowable in the system

is 120°C. Calculate the number of stages that are necessary in the compression and the rate of cooling water circulated per kg of air. Take c_p = 1 kJ/kg. K.

Solution

The $p-V$ diagram for the compressor is shown in Fig. 12.21.

T_1 = 273 +15 = 288 K, p_1 = 1.03 bar, n =1.35

Since the maximum allowable temperature is 120°C, the temperature after first stage of compression,

$$T_2 = 273 + 120 = 393 \text{ K}$$

The intercoolers cool the air from T_2 = 393 K to T_3 = 273 + 45 =318 K. For the second stage, the inlet temperature is T_3 = 318 K and outlet temperature is T_4 = 393 K (see Fig. 12.10b).

Figure 12.21 *p-V diagram*

Pressure ratio

The pressure ratio is same for all the subsequent stage till the pressure reaches 135 bar. Hence, the number of stage i can be calculated from,

$$i \ln 2.26 = \ln 3.952$$
$$i = 4.51$$

Therefore, the minimum number of stages required including the first stage

$$= 4.51 + 1 = 5.51 \sim 6$$

Heat received by stage coolers per kg of air compressed

$$= c_p (T_2 - T_3)(i - 1) = 1(393 - 318) \times 5 = 375 \text{ kJ}$$

Cooling water circulated per kg of air

Example 12.8

A single-acting, two-stage reciprocating compressor with complete intercooling delivers 10 kg/min of air at 16 bar. The suction occurs at 1 bar and 27°C. The compression and expansion processes are reversible with polytropic index $n = 1.3$. The compressor runs at 450 rpm.

Calculate the following:

1. The power required to drive the compressor.
2. The isothermal efficiency.
3. The free air delivered.
4. The heat transferred in intercooler.
5. The swept and clearance volumes for each cylinder if the clearance ratios for L.P. and H.P. cylinders are 0.04 and 0.06, respectively.

Solution

The $p-V$ diagram is shown in Fig. 12.22.

p_1 = bar, p_3 = 16 bar, T_1 = 273 + 27 = 300 K, N = 450 rpm, n = 1.3, c_1 = 0.04, and c_2 = 0.06

For perfect intercooling,

Figure 12.22 *$p-V$ diagram for two-stage compressor*

1. Work done in two stages with perfect intercooling,
2. Isothermal power,
3. Free air delivered

4. Heat transferred in intercooler with perfect

intercooling

5. Clearance volume = c_1 $(V_1 - V_3) = c_1 V_s = 0.04 \times 0.0207$

$$= 8.285 \times 10^{-4}\ \text{m}^3$$

6. **H.P. stage:**

Clearance volume = $c_2 V_s = 0.06 \times 5.4 \times 10^{-3} = 3.24 \times 10^{-4}\ \text{m}^3$

Example 12.9

A reciprocating compressor is to be designed to deliver 4.5 kg/min of air from 100 kPa and 27°C to compress through an overall pressure ratio of 9. The law of compression is $pV^{1.3}$ = constant. Calculate the saving in power consumption and gain in isothermal efficiency, when a two-stage compressor with complete intercooling is used in place of a

single-stage compressor. Assume equal pressure ratio in both the stages of the two-stage compressor. Take R = 0.287 kJ/kg K.

Solution

Given: m = 4.5 kg/min, p_1 = 100 kPa, T_1 = 273 + 27 = 300 K, r_p = 9, n = 1.3

Rate of work required in single-stage compression

Rate of work required in two-stage compression

Saving in power = 5.54 – 4.84 = 0.70 kW

Example 12.10

A double-acting, single-cylinder reciprocating air compressor has a piston displacement of 0.015 m^3 per revolution, operates at 500 rpm, and has 5% clearance ratio. The air is received at 1 bar and delivered at 6 bar. The compression and expansion are polytropic with $n = 1.3$. If the inlet temperature of air is 20°C, determine (a) the volumetric efficiency, (b) the power required and (c) the heat transferred and its direction, during compression.

Solution

Given that $V_s = 0.015\ m^3/rev$, $N =$

500 rpm, $c = V_c/V_s = 0.05$

$p_1 = 1$ bar, $p_2 = 6$ bar, $n = 1.3$, $T_1 = 273 + 20 = 293$ K

1. Volumetric efficiency,
2. Volumetric efficiency

or $V_a = 0.015 \times 0.8516 \times 500 \times 2 = 12.774$ m^3/min

Power required,

= 2834.54 kJ/min or 47.242 kW

3. Mass of air sucked,

Heat transferred during compression,

Example 12.11

Find the optimum intermediate pressure of a two-stage reciprocating compressor, if intercooling is done up to a

temperature , which is greater than the inlet temperature.

Solution

Work done in the L.P. cylinder (Fig. 12.23)

Work done in the H.P. cylinder,

Total work done, $W = W_1 + W_2$

Figure 12.23 *p − V diagram for two-stage compressor*

For maximum work,

Example 12.12

A two-stage reciprocating air

compressor takes in air at 1.013 bar and 15°C and delivers at 43.4 bar. The intercooler pressure is 7.56 bar. The intercooling is perfect and the index of compression is 1.3. Calculate the work done in compressing 1 kg of air. If both cylinders have the same stroke and the piston diameters are 9 cm and 3 cm and the volumetric efficiency of the compressor is 90%, will the intercooler pressure be steady or will rise or fall as the compressor continues working?

Solution

Given that p_1 = 1.013 bar, T_1 = 273 + 15 = 288 K

$p_3 = 43.4$ bar, $p_2 = 7.56$ bar, $n = 1.3$,

$m = 1$ kg, $d_1 = 9$cm, $d_2 = 3$ cm, $\eta_v = 0.9$

Input work per kg for perfect intercooling $T_3 = T_1$,

Compression in L.P. cylinder:

For constant pressure in the intercooler, the volume of air leaving the intercooler and entering the H.P. cylinder,

Ratio of effective cylinder volumes,

As more air is supplied to H.P.

cylinder than it can hold and consequently the pressure in the intercooler will rise.

Example 12.13

Find the percentage saving in work done by compressing air in two stages from 1 bar to 7 bar instead of one stage. Assume compression index 1.35 in both the cases and optimum pressure and complete intercooling in two-stage compression.

Solution

Given that $p_1 = 1$ bar, $p_3 = 7$ bar, $n_1 = n_2 = 1.35$

For perfect intercooling, = 2.646 bar and $T_3 = T_1$

With compression in one stage (see Fig. 12.7) without clearance, work done per kg of air is given by,

Work done with perfect intercooling,

Example 12.14

A single-stage, double-acting reciprocating air compressor delivers 3 m^3 of free air/min at 1.013 bar and 20°C to 8 bar with the following data:

Rotational speed = 300 rpm, mechanical efficiency $\eta_{mech} = 0.9$, pressure loss in passing through intake valve = 0.04 bar, temperature rise of air during suction stroke = 12°C, clearance volume = 5% of stroke volume. Index of compression and expression = 1.35, and length of stroke = 1.2 times of the cylinder diameter. Calculate: (a) volumetric efficiency, (b) cylinder dimension, (c) indicated power, and (d) isothermal efficiency of the compressor, take for air, $R = 0.287$ kJ/kg K.

Solution

Given that $V_a = 3$ m^3/min, $p_a =$ 1.013 bar, $T_a = 273 + 20 = 293$ K,

$p_1 = 1.013 - 0.04 = 0.973$ bar, $T_1 = T_a + 12 = 293 + 12 = 305$ K, $V_c = 0.05V_s$, $n = 1.35$, $L = 1.2D$, $R = 0.287$ kJ/kg K, $N = 300$ rpm

The $p - V$ diagram is shown in Fig 12.24.

Mass of free air delivered,

Compression process 1–2:

Figure 12.24 *$p - V$ diagram for single stage compressor*

Expansion process 3–4:

$$\dot{V}_3 = \dot{V}_c = 0.05\dot{V}_s$$
$$\dot{V}_1 - \dot{V}_4 = (\dot{V}_c + \dot{V}_3) - \dot{V}_4 = (0.05\dot{V}_s + \dot{V}_s) - 0.238\dot{V}_s = 0.812\dot{V}_s$$

Corresponding value of F.A.D. per cycle,

1. Volumetric efficiency,
2. Volume of air inhaled per cycle,

Stroke volume, V_s = = $0.94248D^3$ m^3/cycle

Also stroke volume = = 6.6756×10^{-3} m^3/cycle

$\therefore 0.94248D^3 = 6.6756 \times 10^{-3}$

$D = 0.192$ m or 192 mm

$L = 1.2 \times 192 = 230.4$ mm

3. Indicated power, I.P. = $\dot{m}R\,(T_2 - T_1)$
4. Isothermal indicated power,

Isothermal efficiency,

Example 12.15

The following data were obtained from a performance test of a 14 cm × 10 cm single stage reciprocating air compressor having 3 percent clearance: barometer 77 cm Hg, suction pressure 0 bar gauge,

suction temperature 22°C, discharge pressure 4.05 bar gauge, discharge temperature 174°C, shaft rpm 1160, shaft power 350 kW, mass of air delivered per minute 1.75 kg. Determine (a) the actual volumetric efficiency; (b) the approximate indicated power; (c) the isothermal compression efficiency; (d) the mechanical efficiency and (e) the overall efficiency of the unit. Take p_{atm} = 1.031 bar.

Solution

Atmospheric pressure corresponding to 77 cm Hg

$\therefore p_1 = 0 + 1.0447 = 1.0447$ bar

and $p_2 = 4.05 + 1.0447465 = 5.0947$ bar

Air entering the cylinder

Piston displacement

1. Volumetric efficiency
2. ∴ Compression index $n = 1.353$
3. Isothermal power
4. Compressor mechanical efficiency
5. Overall efficiency = isothermal efficiency × mechanical efficiency

$$= 0.8023 \times 0.836 = 67.07\%$$

Example 12.16

A two-stage reciprocating compressor is used to compress from 1.0 bar to 16 bar. The compression is as per the law $pV^{1.25}$ = const. The temperature of air at

inlet of compressor is 300 K. Neglecting the clearance and assuming perfect intercooling find out the minimum indicated power to deliver 5 m^3/min air measured at inlet conditions and find the intermediate pressure also.

Solution

Given: $p_1 = 1$ bar $v_1 = 5 m^3/min = 0.083\ m^3/s$, $p_3 = 16$ bar, $n = 1.25$, $T_1 = 300$ K

For complete intercooling, the intercooler pressure,

Work done in compressing the air,

Power required to drive the compressor is 26.519 kW.

Summary for Quick Revision

1. A reciprocating compressor is a device in which air is compressed in a conventional cylinder with a closely fitted piston making reciprocating motion.
2. Free air delivered (FAD) is the actual volume delivered by the compressor at the stated pressure, reduced to intake pressure and temperature and expressed in m^3/min.
3. The work done on the compressor to compress a given volume of air at a given pressure shall be the least when the compression process is isothermal.
4. Approximation to isothermal compression can be achieved by spraying cold water on the cylinder, water jacketing the cylinder, intercooling by using multi-stage compression and by using external fins of the cylinder surface.
5. Single stage compression:
    1. Required work and power.
        1. Without clearance:

            Power required in driving the compressor,

            Where N = rpm, if single acting

            = number of strokes, if double acting.
        2. With clearance:
    2. When referred to ambient conditions,

        where = clearance ratio, V_c= clearance

volume, V_s = swept volume.

    3. Isothermal efficiency,
    4. Adiabatic efficiency,
    5. Main dimensions,
6. Two-stage compression:
    1. For perfect intercooling,
    2. Minimum compression work, $(W_t)_{min}$
    3. In general, with i number of stages,
    4. Minimum shaft work,
    5. Minimum indicated power with imperfect intercooling,
    6. Heat rejected to intercooler = $mc_p\,(T_2 - T_3)$
    7. Cylinder dimensions:
    8. Effectiveness of intercooler,
7. Indicated power,
8. Mechanical efficiency,
9. Heat rejected per kg of air,
10. A compressor can be controlled by: Throttle control, clearance control or by blowing air to waste.

Multiple-choice Questions

1. Which one of the following statements is correct? In reciprocating compressors, one should aim at compressing the air
    1. adiabatically
    2. isentropically
    3. isothermally
    4. polytropically
2. Consider the following statements:
    1. Reciprocating compressors are best suited for high pressure and low volume capacity.
    2. The effect of clearance volume on power consumption is negligible of the same volume of discharge.
    3. iii. While the compressor is idling, the delivery value is kept open by the control circuit.
    4. Intercooling of air between the stages of compression helps to minimize losses. Of these statements
    1. i and ii are correct

2. i and iii are correct
3. ii and iv are correct
4. iii alone is correct

3. For two-stage compressor in which index of compression for low pressure stage is m and for high pressure stage in *n*. The load sharing with perfect intercooling in expressed as:
   1.
   2.
   3.
   4.
4. For a two-stage reciprocating compressor, compression from pressure p_1 to p_3 is with perfect intercooling and no pressure losses. If compression in both cylinders follows the same polytropic process and the atmospheric pressure is p_a then the intermediate pressure p_2 is given by
   1. $p_2 = (p_1 + p_3)/2$
   2.
   3.
   4.
5. A large clearance volume in reciprocating compressor results in
   1. reduced volume flow rate
   2. increased volume flow rate
   3. lower suction pressure
   4. lower delivery pressure
6. In a reciprocating air compressor, the compression work per kg of air
   1. increases as clearance volume increases
   2. decreases as clearance volume increases
   3. is independent of clearance volume
   4. increases with clearance volume only of multistage compressor
7. Consider the following statements:

When air is to be compressed to reasonably high pressure, it is usually carried out by multistage compressor with an intercooler between the stages because

1. work supplied is saved.
2. weight of compressor is reduced.
3. more uniform torque is obtained leading to the

reduction in the size of flywheel.
4. volumetric efficiency is increased.

Of the four statements listed above

1. 1 alone is correct
2. 2 and 4 are correct
3. 1, 2 and 3 are correct
4. 1, 2, 3 and 4 are correct

8. Consider the following statements:

   The volumetric efficiency of a compressor depends upon

   1. clearance volume
   2. pressure ratio
   3. index of expansion

   Of these statements:

   1. 1 and 2 are correct
   2. 1 and 3 are correct
   3. 2 and 3 are correct
   4. 1, 2 and 3 are correct
9. The heat rejection by a reciprocating air compressor during the reversible compression process AB, shown in the temperature-entropy diagram, is represented by the area:
   1. ABC
   2. ABDE
   3. ABFG
   4. ABFOE
10. A 3-stage reciprocating compressor has suction pressure of 1 bar and delivery pressure of 27 bar. For minimum work of compression, the delivery pressure of 1st stage is
    1. 14 bar
    2. 9 bar
    3. 5.196 bar
    4. 3 bar
11. Consider the following factors:

    1. Cylinder size

2. Clearance ratio
3. Delivery pressure
4. Compressor shaft power

The factors which affect the volumetric efficiency of a single-stage reciprocating air compressor would include

1. 1 and 2
2. 3 and 4
3. 2 and 3
4. 1 and 4

12. A four-stage compressor with perfect intercooling between stages, compresses air from 1 bar to 16 bar. The optimum pressure in the last intercooler will be
    1. 6 bar
    2. 8 bar
    3. 10 bar
    4. 12 bar
13. The clearance volume of a reciprocating compressor directly affects
    1. piston speed
    2. noise level
    3. volumetric efficiency
    4. temperature of air after compression
14. The capacity of an air compressor is specified as $3m^3$/min. It means that the compressor is capable of
    1. supplying 3 m^3 of compressed air per minute
    2. compressing 3 m^3 of free air per minute
    3. supplying 3 m^3 of compressed air at NTP
    4. compressing 3 m^3 of standard air per minute
15. A two-stage compressor takes in air at 1.1 bars and discharges at 20 bars. For minimum work input, the intermediate pressure is
    1. 10.55 bars
    2. 7.33 bars
    3. 5.5 bars
    4. 4.7 bars
16. Consider the following statements:

The volumetric efficiency of a reciprocating

compressor can be enhanced by

1. heating the intake air
2. decreasing the clearance volume
3. cooling the intake air

Which of these statements is/are correct?

1. 1 alone
2. 1 and 2
3. 2 and 3
4. 3 alone

17. Reciprocating compressors are provided with
    1. simple disc/plate valve
    2. poppet valve
    3. spring-loaded disc value
    4. solenoid valve
18. If n is the polytropic index of compression and is the pressure ratio for a three-stage compressor with ideal intercooling, the expression for total work of three stage is
    1.
    2.
    3.
    4.
19. Consider the following statements:

Volumetric efficiency of a reciprocating air compressor increases with

1. increase in clearance ratio
2. decrease in delivery pressure
3. multistaging

Which of the statements given above is/are correct?

1. Only 1 and 2
2. Only 2 and 3
3. Only 3

4. 1, 2 and 3

20. What is the preferred intercooler pressure for a two stage air compressor working between the suction pressure p_s and the delivery pressure p_d?
    1.
    2.
    3.
    4.
21. Which of the following statements are correct for multi-staging in a reciprocating air compressor?

    1. It decreases the volumetric efficiency
    2. The work done can be reduced
    3. A small high-pressure cylinder is required.
    4. The size of flywheel is reduced.

    Select the correct answer using the code given below:

    1. 1, 2 and 3
    2. 2, 3 and 4
    3. 1, 3 and 4
    4. 1, 2 and 4
22. For a two-stage reciprocating air compressor, the suction pressure is 1.5 bar and the delivery pressure is 54 bar. What is the value of the ideal intercooler pressure?
    1. 6 bar
    2. 9 bar
    3. 27.75 bar
    4.
23. Consider the following statements:

    In a reciprocating compressor, clearance volume is provided

    1. so that piston does not hit and damage the valves
    2. to account for differential thermal expansion of piston and cylinder
    3. to account for machining tolerances
    4. to achieve isentropic compression

Which of these statements are correct?

1. 1, 2 and 3
2. 1, 2 and 4
3. 1, 3 and 4
4. 2, 3 and 4

24. Which of the following can be the cause/causes of an air-cooled compressor getting overheated during operation?

    1. Insufficient lubricating oil.
    2. Broken valve strip.
    3. Clogged intake filter.

Select the correct answer using the code given below:

1. Only 3
2. Only 1 and 2
3. Only 2 and 3
4. 1, 2 and 3

25. Performance of a reciprocating compressor is expressed by
    1.
    2.
    3.
    4.

Explanatory Notes

1. 10. (d)
2. 12. (b) Optimum pressure in the last intercooler
3. 15. (d)
4. 22. (b)

Review Questions

1. List at least six uses of compressed air in industry.
2. Explain the working of a single stage reciprocating air compressor.

3. What do you mean by free air delivered (FAD)?
4. Define volumetric efficiency.
5. What are the factors on which volumetric efficiency depends?
6. What are the methods generally adopted for approximating the compression process of a reciprocating air compressor as isothermal one?
7. Define clearance ratio. How the volumetric efficiency is dependent on clearance ratio?
8. Define isothermal efficiency and compressor efficiency.
9. Differentiate between isothermal power and indicated power.
10. Define adiabatic efficiency.
11. What are the advantages of multi-stage compression?
12. What is the condition for minimum compressor work with perfect intercooling for a multi-stage reciprocating air compressor?
13. Distinguish between the function of intercooler and after-cooler.
14. What is an air motor?
15. How heat is rejected in air compressor?

Exercises

12.1 A single-stage single-acting reciprocating air compressor delivers 15 m^3 of free air per minute from 1 bar to 8 bar at 300 rpm.The index of both compression and expansion is $n = 1.3$ for and clearance of swept volume, find the diameter and stroke of the compressor. Take $L = 1.5\ D$.

[**Ans.** 383 mm, 587.5 mm]

12.2 A single-stage double-acting reciprocating air compressor delivers air at 7 bar. The pressure and temperature at the end of suction stroke are 1 bar and 27°C. It delivers 2 m^3 of free air per minute when the compressor is running at 300 rpm. The clearance volume is 5% of the stroke volume. The ambient pressure and temperature are 1.03 bar and 20°C. Index of compression and expansion is 1.30 and 1.35 respectively. Calculate (a) the volumetric efficiency, (b) indicated power and brake power if mechanical efficiency is 80%, and (c) diameter and stroke of the cylinder if both are equal.

[**Ans.** 83.9%, 8.5 kW, 10.64 kW, 174.6 mm]

12.3 A two-stage single-acting reciprocating air compressor delivers air at 20 bar. The pressure and temperature of air before compression in L.P. cylinder are 1 bar and 27°C. The discharge pressure of L.P. cylinder is 4.7 bar. The pressure of air leaving the intercooler is 4.5 bar and the air is cooled to 27°C. The diameter and stroke of L.P. cylinder are 0.4 m and 0.5 m respectively. The clearance volume is 4% of stroke in both cylinders. The speed of compressor is 200 rpm. The index of compression and re-expansion in both cylinders is 1.3. Determine (a) the indicated power to run the compressor, and (b) the heat rejected to intercooler per minute.

[**Ans.** 68.9 kW, 1716 kJ/min]

12.4 A single-stage, double-acting reciprocating air compressor delivers 15 m^3 of air per minute measured at 1.013 bar, 27°C and delivers at 7 bar. At the end of the suction stroke the pressure and temperature are 0.98 bar and 40°C. The clearance volume is 4% of the swept volume and the stroke is 1.3 times the bore. The compressor runs at 300 rpm. Calculate (a) the volumetric efficiency, (b) cylinder dimensions, (c) indicated power and (d) isothermal efficiency. $n = 1.3$ for both compression and expansion. Take $R = 0.287$ kJ/kg.K

[**Ans.** 79.6%; 313.3 mm, 407.3 mm; 65.83 kW; 78.92%]

12.5 A single-stage, double acting air reciprocating compressor takes air at 0.98 bar and 32°C and delivers at 6.32

bar. The clearance is 5% of the stroke volume. The compression and expansion follow the law $pV^{1.32} = c$. The air handled by the compressor is 17 m^3/min when measured at 1 bar and 15°C. Determine the temperature of air delivered, the stroke volume and the indicated power of compressor if it turns at 500 rpm. Neglect the area of piston and take $R = 0.287$ kJ/kg.K.

12.6 The free air delivered of a single cylinder single stage reciprocating air compressor is 2.5 m^3/min. The ambient air is at 0°C and 1.013 bar and delivery pressure is 7 bar. The clearance ratio is 5% and law of compression and expansion is $pv^{1.25} = c$. If $L = 1.2\ D$ and the compressor runs at 150 rpm,

determine the size of the cylinder.

[**Ans.** 275.5 mm, 330 mm]

12.7 A single stage double acting reciprocating air compressor delivers air at 7 bar. The amount of free air delivered is 2 m^3 at 300 rpm. The pressure and temperature at the end of suction stroke are 1 bar and 27°C. The ambient conditions are 1.03 bar and 20°C. The clearance is 5% stroke. The compression and re-expansion follow the law $pV^{1.3}$ = const. Determine the brake power required to run the compressor if the mechanical efficiency is 80%, and the diameter and stroke of the cylinder if both are equal.

[**Ans.** 10.8kW, 179.5 mm]

12.8 A two-stage double acting

reciprocating air compressor delivers air at the rate of 1.35 kg/s. The suction pressure is 1 bar and inter-stage pressure is 7 bar and delivery pressure 42 bar. Air enters the L.P. cylinder at 17°C and cooled in the intercooler to 32°C. The clearances in L.P. and H.P. cylinders are 6% and 8% of respective strokes. The law of compression and re-expansion is $pV^{1.21} = c$ in both cylinders. The compressor runs at 500 rpm. Calculate the amount of cooling water required per minute in intercooler, if rise in temperature of water is limited to 20°C, power required, and size of cylinder if length of stroke and bore are equal.

[**Ans.** 2259 kg/min, 1068 kW, 483 mm]

12.9 Determine the size of a cylinder for

a double acting reciprocating air compressor of 37 kW, in which air is drawn in at 1 bar, 15°C and compressed according to the law $pV^{1.2}$ = const. to 6 bar. The compressor runs at 100 rpm with average piston speed of 152.5 m/min. Neglect clearance.

[**Ans.** 298.5 mm, 762.5 mm]

12.10 A single stage double acting reciprocating air compressor running at 500 rpm handles 17 m^3/min of air, measured at 1 bar and 15°C. The pressure and temperature at the end of suction are 0.98 bar and 32°C. The air is delivered at 6.325 bar. Assuming a clearance factor of 5% and the compression and expansion processes to follow the law $pV^{1.32}$ = const, determine

the stroke volume of the compressor. Also calculate the indicated power of the compressor. Take $R = 0.287$ kJ/kg.K

[**Ans.** 0.2175 m^3, 70.73 kW]

12.11 A single acting compressed air motor works on compressed air at 10.5 bar and 37°C supplied at the rate of 1 kg/min. Cut-off takes place at 25% of the stroke and the expansion follows the adiabatic frictionless law down to 1.0135 bar. Determine the mean effective pressure, the indicated power and the cylinder volume if the motor runs at 250 rpm. Neglect clearance.

[**Ans.** 3.37 bar, 1.94 kW, 0.001383 m^3]

12.12 The cylinder of an air motor has a bore of 63.5 mm and a stroke of 114 mm. The supply pressure and

temperature are 6.3 bar and 24°C and exhaust pressure is 1.013 bar. The clearance volume is 5% of the swept volume and the cut-off ratio is 0.5. The air is compressed by the returning piston after it has travelled through 0.95 of its stroke. The law of compression and expansion is $pV^{1.3}$ = const. Calculate the temperature at the end of expansion and the indicated power of the motor which runs at 300 rpm. Also calculate the air supplied per minute. Take R = 0.287 kJ/kgK.

[**Ans.** 244.4 K, 0.746 kW, 0.418 kg/min]

12.13 A water cooled air compressor requires a work input of 200 kJ/kg of air delivered. The enthalpy of air leaving the compressor is 75 kJ/kg greater than

that entering. Heat lost to the cooling water is 105 kJ/kg. From the first law analysis estimate the heat lost by compressor to atmosphere.

12.14 A two-stage single acting air reciprocating compressor equipped with an intercooler draws air at 1.013 bar and 30°C and delivers it at 20 bar. The mass of air delivered by the compressor is 10 kg/min. The ratio of clearance volume to swept volume in the low pressure and high pressure cylinders is 0.03 and 0.04 respectively. If the compressor has a mechanical efficiency of 82% and it operates under most efficient conditions, determine:

1. power supplied at the shaft of the compressor;
2. isothermal efficiency;
3. ratio of cylinder diameters for identical stroke length; and
4. heat rejected in the intercooler.

Take $R = 0.287$ kJ/kg.K, $c_p = 1.005$ kJ/kgK and $\gamma = 1.4$.

12.15 A single acting two-stage reciprocating air compressor is to compress air from 1 bar and 30°C to 12 bar. The base of the low pressure cylinder is 30 cm. The stroke length of both the low and the high pressure cylinders is the same and is equal to 40 cm. The compressor runs at 180 rpm. The clearance volume in both the cylinders is 3% of the stroke volume. Index of compression and expansion is 1.3 in both the cylinders. Determine the shaft power required to drive the compressor when (a) the air is cooled to its original temperature before entering the H.P. cylinder, (ii) when the air is

cooled to 45°C in the intercooler. Assume mechanical efficiency to be 85% in both cases. Take R = 0.287 kJ/kg.K.

12.16 A multi-stage reciprocating compressor has to be designed to supply air at 135 bar, while atmospheric condition is 1.03 bar and 15°C. The value of compression index may be assumed as 1.35. Due to practical reasons the intercoolers are not able to cool the air below 45°C, while the maximum temperature allowable in the system is 120°C. Calculate the number of stages that are necessary in the compression and the rate of cooling water circulated per kg of air. Take c_p = 1 kJ/kgK.

ANSWERS TO MULTIPLE-CHOICE QUESTIONS

1. c
2. b
3. a
4. b
5. d
6. a
7. a
8. d
9. b
10. d
11. a
12. b
13. c
14. b
15. d
16. c
17. a
18. d
19. d
20. c
21. b
22. b
23. a
24. c
25. a

Chapter 13

Rotary Air Compressors

13.1 ❒ INTRODUCTION

In a rotary compressor, the compression of air is achieved due to the rotating blades fitted in a rotor. It requires less starting torque as compared to reciprocating compressors because of direct coupling with the prime over. Usually, rotary compressors operate at high speed and supplies higher quantity of air than reciprocating compressors. Rotary compressors may be classified as: (a) Roots blower, (b) Vane type compressor, (c) Lysholm compressor and (d) Screw compressor.

13.2 ❐ WORKING PRINCIPLE OF DIFFERENT ROTARY COMPRESSORS

In positive displacement type rotary compressors, the air is compressed by entrapping it between the reduced space of two sets of engaging surfaces. The pressure rise is either by backflow of air, as in the case of roots blower, or by both squeezing action and backflow of air, as in the case of vane type.

13.2.1 Roots Blower or Lobe Compressor

The schematic arrangement of a roots blower is shown in Fig. 13.1(a). It consists of two rotors that are driven externally. One rotor is connected to the driver and the second rotor is gear driven from the first in the opposite direction. The lobes of the rotors are of epicycloids, hypocycloid, or involute

profiles to ensure a seal between the high and low pressure regions at all angular positions. A small clearance provided between the rotors and the cylinder surface to reduce the wear reduces the efficiency of the compressor due to leakage.

The volume V_s of air at atmospheric pressure p_1 is entrapped between the left hand rotor and the cylinder casing. The volume of air once entrapped does not decrease from entry to exit, and therefore, there is no rise of pressure till the exit port is uncovered. As the exit part opens, some high pressure air from the receiver rushes back and mixes with the air volume V_s irreversibly until the pressure is equalised. The resulting

pressure after mixing will be the receiver pressure p_2. The air is then transferred to the receiver. This happens four times in one revolution in case of two-lobed rotor and six times in case of three-lobed rotor. The $p-V$ diagram for the roots blower is shown in Fig. 13.1(b).

Figure 13.1 *Roots blower*

where i = number of lobes

Let V_1 = volume of air handled per minute at p_1 and T_1.

Then work done per minute is,

If the compression is isentropic, then ideal work required per minute is,

The efficiency of roots blower

where r_p = is the pressure ratio.

The roots blower is used to supply air from 0.15 to 1500 m^3/min with pressure ratio up to 3.6 per blower. Rotational speeds up to 12,500 rpm are used and can be directly coupled to a steam turbine or a gas turbine shaft without any intermediate gearing.

Figure 13.2 *Vane type compressor*

13.2.2 Vanes Type Blower

The schematic diagram of a vane type blower is shown in Fig. 13.2(a). It consists of a rotor located eccentrically in a cylindrical outer casing. The rotor

carries a set of spring-loaded vanes in the slots of the rotor. The air of volume V_1 at atmospheric pressure p_1 is entrapped between two vanes. As the rotation proceeds counter clockwise, the entrapped air is first compressed reversibly to V_a. Afterwards, the compression continues to V_c (say) and then to V_d. After this, the delivery port is uncovered and the irreversible compression takes place from pressure p_d to p_2. The p–V diagram for the compression process is shown in Fig. 13.2(b). The work done per revolution with i vanes is given by:

The vane blowers require less power than the root blowers for the same capacity and pressure rise. They are

used to deliver up to 150 m^3/min of air at pressure ratio up to 8.5. The speed is limited to 3000 rpm.

13.2.3 Lysholm Compressor

The Lysholm compressor is shown in Fig. 13.3. Its principle of working is similar to that of the roots blower. The air is admitted through one end of the compressor and trapped between the helical rotors and the casing. The screw action of the rotors displaces the air axially. It produces constant compression internally. Its main disadvantage is mechanical complexity.

Figure 13.3 *Lysholm compressor*

13.2.4 Screw Compressor

The screw compressor may be single

helical or double helical. The advantage of double helical is that they are balanced axially. The air is carried forward to the discharge along the rotor in pockets formed between the teeth and the casing as shown in Fig. 13.4. The principle of working is similar to that of the roots blower.

Figure 13.4 *Screw compressor*

Example 13.1

Free air of 25 m^3/min is compressed from 1 bar to 2.5 bar. Calculate the indicated power required if the compression is carried out in (a) roots blower and (b) vanes blower. Assume that there is 20% reduction in volume before the backflow

occurs. Also, calculate the isentropic efficiency in each case.

Solution

The p–V diagrams for roots blower and vanes blower are shown in Fig. 13.5(a) and (b), respectively.

Figure 13.5 *p–V diagrams: (a) Roots blower, (b) Vanes blower*

1. Roots blower,
2. Vanes blower,

I.P. if whole compression is carried out isentropically,

Isentropic efficiency of roots blower

Isentropic efficiency of vanes blower

Example 13.2

A rotary vanes blower works between the pressure limits of 1 bar and 1.8 bar, and gives 5 m^3/min of free air delivered when running at 240 rpm. Determine the power required to drive the blower when (a) ports are so placed that there is no internal compression and (b) when the ports are so placed that there is 50% pressure rise due to internal adiabatic compression before backflow occurs. Assuming mechanical efficiency to be 98%, calculate the blower efficiency.

Solution

The *p*–*V* diagrams without and with internal compression are shown in Fig. 13.6(a) and (b), respectively.

Figure 13.6 *p–V diagrams: (a) Without internal compression, (b) With internal compression*

1. Without internal compression,
2. With internal compression,

For minimum power requirement, the whole compression should be a reversible adiabatic process.

$$\text{Actual I.P.} = \text{Ideal I.P.} \times \eta_{\text{mech}}$$

$$= 5.333 \times 0.98 = 5.226 \text{ kW}$$

Isentropic efficiency without internal compression =

Isentropic efficiency with internal compression =

13.3 ❐ COMPARISON OF ROTARY AND RECIPROCATING COMPRESSORS

The comparison of rotary and reciprocating compressors is given in Table 13.1.

Table 13.1 *Comparison of rotary and reciprocating compressors*

Example 13.3

Free air of 30 m^3/min is compressed from 101.3 kPa to 2.23 bar in roots blower. Determine the power required and the isentropic efficiency.

Solution

The $p - V$ diagrams for roots blower is shown in Fig. 13.7.

Figure 13.7 *$p - V$ diagrams*

$V_1 = 30$ m^3/min, $p_1 = 101.3$ kPa
$p_2 = 2.23$ bar

Refer Fig. 13.8.

Indicated power, I.P. =

$$\text{Isentropic indicated power, } (\text{I.P.})_{\text{isen}} =$$

$$\text{Isentropic efficiency, } \eta_{\text{isen}} =$$

Summary for Quick Revision

1. Rotary compressors are of positive displacement type in which air is compressed by entrapping it between the reduced space of two sets of engaging surfaces.
2. The pressure rise is either by backflow of air (as in roots blower) or by both squeezing action and backflow (as in vanes blower).
3. Roots blower

   $$\text{Work done per minute, } W_{\text{roots}} = (p_2 - p_1)V_1$$

   $$\text{Roots efficiency, } \eta_{\text{roots}} =$$

   where V_1 = volume of air handled per minute

   r_p = pressure ratio

4. Vanes type blower

   $$\text{Work done per revolution} =$$

   where i = number of Vanes

p_d = maximum isentropic pressure

5. Indicated power, I.P. = kW

Multiple-choice Questions

1. The efficiency of a roots air blower is (r_p = pressure ratio)
    1.
    2.
    3.
    4.
2. In a roots blower with two lobes, the high pressure air is delivered in one revolution
    1. Once
    2. Twice
    3. Thrice
    4. Four times
3. The efficiency of vane type air compressor as compared to roots blower for the same pressure ratio is
    1. Less
    2. More
    3. Same
    4. May be less or more
4. The $p–V$ diagram shown in Fig. 13.8 below is for

Figure 13.8 *p-V diagram*

    1. Roots blower
    2. Vanes blower
    3. Centrifugal compressor
    4. axial compressor
5. Roots blower is an example of
    1. Reciprocating (positive displacement) compressor
    2. Rotary (positive displacement) compressor
    3. Centrifugal compressor
    4. Axial compressor

Review Questions

1. What are the types of rotary compressors?

2. Draw the $p-V$ diagram for a roots blower.
3. Which rotary compressor is used for steam/gas turbines?
4. What do you mean by a positive displacement compressor?

Exercises

13.1 What are the various types of compressors?

13.2 What is a positive displacement compressor?

13.3 Explain the working of a roots blower.

13.4 Derive an expression for the work done and efficiency of a roots blower.

13.5 Explain the working of a vane type blower.

13.6 Explain the working of a screw compressor.

13.7 Compare rotary and reciprocating compressors.

ANSWERS TO MULTIPLE-CHOICE QUESTIONS

1. a
2. d
3. b
4. b
5. b

Chapter 14

Centrifugal Air Compressors

14.1 ❐ INTRODUCTION

A centrifugal compressor is of rotodynamic type in which air flows continuously and steadily through various parts and the rise in pressure is primarily due to the centrifugal action. It is used to supply large quantities of air but at a lower pressure ratio.

14.2 ❐ CONSTRUCTIONAL FEATURES

Essentially, a centrifugal compressor consists of four elements namely, inlet buckets, impeller, diffuser, and casing as shown in Fig. 14.1. The inlet buckets are

attached to the shaft and rotate with it, which guide air on the impeller. The impeller consists of a disc on which radial blades are attached. The diffuser surrounds the impeller and provides diverging passages for air flow, thus increasing the air pressure. The air coming out from the diffuser is collected in the casing and taken out from the outlet of the compressor.

Figure 14.1 *Centrifugal compressor*

14.3 ❒ WORKING PRINCIPLE

The air enters the eye of the compressor at atmospheric pressure and low velocity. The inlet buckets guide the air to the impeller where it moves radially outward and is guided by the impeller blades. The impeller increases the

momentum of the air flowing through it, causing a rise in pressure and temperature of the air. The air leaving the impeller enters the diffuser where its velocity is reduced by providing more cross-sectional area for the flow of air. A part of the kinetic energy of air is converted into pressure energy and further increases the pressure of air.

14.4 ❒ VARIATION OF VELOCITY AND PRESSURE

As the air flows through the impeller and diffuser, there is a variation of both velocity and pressure as shown in Fig. 14.2. Nearly half of the total pressure rise takes place in the impeller and the remaining half occurs in the diffuser. A pressure ratio of 4 can be achieved in single-stage compressors. For higher

pressure ratios, multi-stage compressors are used. A pressure ratio of 12:1 is possible with multi-stage compressors.

Figure 14.2 *Variation of velocity and pressure of air passing through impeller and diffuser*

14.5 ❒ TYPES OF IMPELLERS

Impellers are of two types—single-eye type and double-eyed type as shown in Fig. 14.3 (a) and (b), respectively. In a single-eye type, air enters into the compressor from one side only, whereas in a double-eyed type, air enters from both sides. A double-eyed type impeller sucks in more air and has the advantage of self-balancing over a single-eye impeller.

In multi-stage compressors, the output of the first stage is passed on to the

second stage and so on, as shown in Fig. 14.3(c).

Figure 14.3 *Types of impellers: (a) Single eye impellers, (b) Double eye impellers, (c) Multi-stage impellers*

14.6 ❒ COMPARISON OF CENTRIFUGAL AND RECIPROCATING COMPRESSORS

The comparison of centrifugal and reciprocating compressors is given in Table 14.1.

Table 14.1 *Comparison of centrifugal and reciprocating compressors*

14.7 ❒ COMPARISON OF CENTRIFUGAL AND ROTARY COMPRESSORS

The comparison of centrifugal and rotary compressors is given in Table 14.2.

Table 14.2 *Comparison of centrifugal and rotary compressors*

14.8 ❒ STATIC AND STAGNATION PROPERTIES

The velocities of air encountered in centrifugal compressors are very high as compared to that in reciprocating compressors. Therefore, total head quantities should be taken into account in the analysis of centrifugal compressors. The total head quantities account for the kinetic energy of the air passing through the compressor.

Consider a horizontal passage of varying area of cross-section, as shown in Fig. 14.4, through which air flows from left to right. Assuming no external heat transfer and work transfer to the system, the steady flow energy equation for one kg mass of air flow can be

written as:

Temperature T is called the 'static temperature'; it is the temperature of the air measured by a thermometer moving with the air velocity. If the moving air is brought to rest under reversible adiabatic conditions, the total kinetic energy of the air is converted into thermal energy, thereby increasing its temperature and pressure. This temperature and pressure of the air are known as 'stagnation' or 'total head' temperature and pressure. The stagnation quantities are denoted by a suffix notation *o*.

Figure 14.4 *Passage of air through varying area of cross-section*

where T_0 = stagnation or total head

temperature

where h_0 = stagnation or total head enthalpy

The stagnation pressure can be obtained by using the following equation:

where p, T = static pressure and temperature

p_0, T_0 = stagnation pressure and temperature.

14.9 ❐ ADIABATIC AND ISENTROPIC PROCESSES

For an adiabatic compression process, there is no heat exchange with the surroundings. If the adiabatic process is reversible (frictionless), then the process

is called isentropic process in which the entropy of the system does not change. However, in an actual compressor, during adiabatic compression, there are losses due to friction in air and blade passages, eddies formation, and shocks at entry and exit. These factors cause internal heat generation and consequently, the maximum temperature reached would be higher than that for adiabatic compression. This results in a progressive increase in entropy. Such a process, although adiabatic, is not reversible adiabatic or isentropic.

The isentropic and adiabatic processes for the static and stagnation values are shown in Fig. 14.5 on the *T*-s diagram. Process 1-2 is the isentropic process for

the static temperature and 01-02 for the total head temperature. Processes 1-2′ and 01-02′ are the adiabatic processes for the static and total head temperatures, respectively.

Figure 14.5 *Isentropic and actual compression*

14.9.1 Isentropic Efficiency

Isentropic efficiency may be defined as the ratio of isentropic temperature rise to actual temperature rise.

where

During compression process, work has to be done on the impeller. The energy balance then gives,

$$c_pT_{01} = c_pT_{02} - W$$

14.10 ❒ VELOCITY DIAGRAMS

The velocity diagrams at inlet and outlet of the impeller for the centrifugal compressor are shown in Fig. 14.6(a).

Figure 14.6 *(a) Actual velocity diagrams*

Let

u_1 = mean blade velocity at entrance

u_2 = mean blade velocity at exit

v_{a1} = absolute velocity at inlet to rotor

v_{a2} = absolute velocity at outlet to rotor

v_{r1} = relative velocity at inlet to rotor

v_{r2} = relative velocity at outlet to rotor

v_{w1} = velocity of whirl at inlet

v_{w2} = velocity of whirl at outlet

v_{f1} = velocity of flow at inlet

v_{f2} = velocity of flow at inlet

D_1, D_2 = mean diameter of rotor at inlet and outlet, respectively

α_1 = exit angle from stator or guide vanes at entrance

β_1 = inlet angle to rotor in impeller blade angle at inlet

α_2 = inlet angle to the diffuser or the stator

β_2 = outlet angle from rotor or impeller

blade angle at outlet.

If it is assumed that the entry of the air to the rotor is axial, then whirl component $v_{w1} = 0$ and $v_{a1} = v_{f1}$.

14.10.1 Theory of Operation

1. Ideal Velocity Diagrams

By Newton's second law of motion, the rate of change of angular momentum of air is equal to the torque applied to the body causing that change. Considering 1 kg/s of mass flow rate of air,

where

For ideal case, let us assume that the impeller is radial vaned, and there is no frictional loss, no heat transfer and the air leaves the impeller with a tangential velocity $v_{w2} = u_2$. Also for axial entry, $v_{w1} = 0$

Work done by impeller for 1 kg/s air flow rate,

Figure 14.6 *(b) Ideal velocity diagrams*

where ω = angular speed of rotor, rad/s

Now, $r_2\omega = u_2$

Since air cannot leave the impeller at a velocity greater than the impeller tip velocity, Eq. (14.11) gives the maximum work capacity of the impeller.

Also, by steady flow energy equation, we have

In most practical problems,

$$v_{a1} = v_{a2}$$
$$= h_2 - h_1$$

where = stagnation pressure ratio

where, = static pressure ratio

Comparing Eq. (14.11) and (14.14), we have

If $v_{a1} = v_{a2}$, then

Total temperature increase across an impeller,

where $\dot{m}$ = mass flow rate of air, kg/s, and u_2 is in m/s.

Therefore, for a centrifugal compressor working under ideal condition, the power input depends on the following:

1. The mass flow rate of air
2. The total pressure ratio of the compressor which, in turn, depends on the square of the impeller velocity
3. The total inlet temperature
4. The total inlet temperature rise between the inlet and outlet
5. There is a maximum work capacity of an impeller depending on the tip velocity

The maximum pressure produced depends on the following:

1. Inlet temperature
2. Square of the impeller velocity
3. It is independent of the impeller diameter

2. Actual Velocity Diagrams

The actual velocity diagrams are shown in Fig. 14.7.

Theoretical torque, $T = v_{w2}r_2 - v_{w1}r_1$

Work done on 1 kg/s of air.

$$w = (v_{w2}r_2 - v_{w1}r_1)w$$

where, ω = angular speed of rotor, rad/s

Figure 14.7 *Actual velocity diagrams for centrifugal compressor*

$= \Delta KE + \Delta p$ due to diffusion action $+ \Delta p$ due to centrifugal action

Therefore, the fraction of kinetic energy imparted to air and converted into pressure energy in impeller is given by,

where ρ = density of air.

If v_{a1} = diffuser outlet velocity, then

14.10.2 Width of Blades of Impeller and Diffuser

Air mass flow rate per second,

where, v_s = specific volume of air

b_1, b_2 = width or height of impeller blades at inlet and outlet, respectively

k_b = blade factor

n = number of blades

t = thickness of blades

The width or height of the diffuser blades at the outlet is given by,

where suffix '*d*' represents the quantities at the diffuser outlet. The width or height of the diffuser blades at the inlet is the same as that of the impeller blades at outlet.

14.11 ❐ SLIP FACTOR AND PRESSURE COEFFICIENT

In the ideal velocity diagrams, we had assumed that $v_{w2} = u_2$. However, in actual operation, this condition is not satisfied in actual practice due to secondary flow effects. Actually, $v_{w2} < u_2$. The difference $(u_2 - v_{w2})$ is called the slip. The slip factor is defined as the ratio of actual whirl component to the ideal whirl component and is denoted by ϕ_s.

The total head pressure ratio is given by,

Substituting the value of (T_{02} - T_{01}) from Eq. (10.5), we have

The actual work done per kg of air is given by:

$$c_p\,(T_{02'} - T_{01}) = u_2 v_{w2}$$

The actual work done per kg of air by the compressor is always greater than $u_2 v_{w2}$ due to fluid friction and windage losses. Therefore, the actual work is obtained by multiplying $u_2 v_{w2}$ by the factor ϕ_w, called the *work factor* or *power input factor*.

$$c_p\,(T_{02'} - T_{01}) = \phi_w u_2 v_{w2}$$

Eq. (10.29) becomes

Now, $v_{\omega 2} = \phi_s u_2$

The *pressure coefficient* is defined as the ratio of isentropic work to Euler work and is denoted by ϕ_p.

Using Eq. (14.5) and assuming radial vanes of impeller, i.e., $v_{w2} = u_2$, we get

Using Eqs (14.28) and (14.30), we get

14.12 ❐ LOSSES

There are two types of losses in a centrifugal compressor: internal and external. The internal losses sustained in the compressor are manifested by an increase of enthalpy of air. The internal

losses which occur in compressor are as follows:

1. Friction between air and walls of flow passage
2. Disc friction
3. Leakage between impeller and casing
4. Turbulence
5. Shock

The compressor sustains an external loss in the form of bearing friction and windage loss. The friction losses are proportional to and hence proportional to . The incidence loss in terms of drag coefficient C_D is proportional to

14.13 ❐ EFFECT OF IMPELLER BLADE SHAPE ON PERFORMANCE

The following types of blade shapes are used for impellers of centrifugal compressors:

1. Backward-curved blades ($\beta_2 < 90°$)
2. Radial blades ($\beta_2 = 90°$)
3. Forward-curved blades ($\beta_2 > 90°$).

The characteristics of these blades are shown in Fig. 14.8. Centrifugal effects on the curved blades create a bending moment and produce increased stresses which limit the maximum speed at which the impeller can run.

Slightly backward-curved impeller blades (i.e., $\beta_2 < 90°$) give optimum efficiency because the degree of reaction increases with decreasing blade tip angles. With increase in degree of reaction, the part of kinetic energy that is transferred into pressure energy within the impeller is also increased. By selecting very small tip blade angles β_2, the length of impeller channel increases and consequently the friction loss increases considerably, thereby reducing

the efficiency. The optimum blade angle is about 45°.

Figure 14.8 *Characteristics of blades*

Radial blades are used for aircraft centrifugal compressors because they can be manufactured easily, have the lowest centrifugal stresses, are free from bending stresses, and have equal energy conversion in impeller and diffuser, giving higher pressure ratio and efficiency.

14.14 ❐ DIFFUSER

The diffuser converts the kinetic energy imparted to air by the impeller into pressure rise. In the vaned diffuser, the vanes are used to remove the whirl of the fluid at a higher rate than is possible by a simple increase in radius, thereby

reducing the length of flow path and diameter. A typical vaned type diffuser is shown in Fig. 14.9. A pressure gradient exists in the diffuser in the direction opposite to that of flow, and the flow streamlines will tend to break away from the diverging passage walls, reversing their direction and resulting in turbulence. About 20° is accepted as the maximum included angle of divergence for satisfactory diffusion.

The clearance between the impeller and the vanes leading edges constitutes a vaneless diffuser. This helps smoothen out velocity variation between the impeller tip and diffuser vanes. This also reduces the circumferential pressure gradient at the impeller tip.

The flow follows an approximately logarithmic spiral path to the vanes after which it is constrained by the diffuser channels.

Figure 14.9 *Vaned diffuser*

where suffices 3 and 4 denote upstream and downstream conditions of diffuser and ρ is the mass density of air.

14.15 ❐ PRE-WHIRL

In order to avoid acceleration to sonic or supersonic velocities, the Mach number of flow entering the impeller eye should be kept below the value of 0.9. When the absolute velocity of approach is high enough or the static absolute temperature of the entering air is low enough to make the Mach number more

than 0.9, the method of providing a pre-whirl and thereby reducing the relative velocity entering the impeller eye is quite helpful.

Figure 14.10 *Effect of pre-whirl on inlet velocity triangle: (a) With pre-whirl ($v_w \neq 0$), (b) Without pre-whirl ($v_w = 0$)*

Pre-whirl is provided by intake guide vane as shown in Fig. 14.10, which shows the velocity triangles for blades without pre-whirl and with pre-whirl. It may be observed that the magnitude of the relative velocity with pre-whirl vane gets reduced at the entrance, but the flow enters with a large relative angle. Thus, the flow inlet Mach number is lowered by pre-whirl vane. However, since $v_{w1} \neq 0$ with pre-whirl vane, the work done on the air must be greater for the same compressor inlet and outlet

conditions.

14.16 ❐ PERFORMANCE CHARACTERISTICS

Let us consider the following variables for the performance of the compressor.

p_3 = pressure at the outlet of diffuser

p_1 = inlet pressure

T_1 = inlet temperature

D = diameter

G = mass rate of flow

N = r.p.m of rotor

Then, $p_3 = f(p_1, T_1, D, G, N)$

The following dimensions are chosen:

M = mass (kg), L = length (m), t = time (s)

Number of variables, $n = 6$

Basic dimensions, $m = 3$

According to Buckingham's *Pi* theorem, the number of dimensionless groupsm = $n - m = 6 - 3 = 3$

The three dimensionless groups are formed by combining p_1, T_1, and D with each of the remaining variables, so that

Substituting the basic dimensions for the variables in π_1, we have

$$\pi_1 = (M^1L^{-1}t^{-2})^{a_1} (L^2t^{-2})^{b_1} L^{c_1}(M^2L^{-1}t^{-2})^1$$

For M, $0 = a_1 + 1$

For L, $0 = - a_1 + 2b_1 + c_1 -1$

For t, $0 = - a_1 - 2b_1 -2$

From which, $a_1 = -1$, $b_1 = 0$, $c_1 = 0$

For a given compressor under consideration, D = const.

Figure 14.11

Similarly, the ratio can be formed to be function of the dimensionless groups of Eqs. (14.35) and (14.36). Similarly, if stagnation temperature and pressure are considered, the result will again be the same except that stagnation

temperatures and pressures are substituted in place of static pressures and temperatures.

Figure 14.12 *Pressure ratio v's Volume flow rate*

The performance of a given compressor or geometrically similar compressor can be plotted in terms of three dimensionless parameters provided dynamic similarity is maintained. Fig. 14.11 shows a plot of against and . The operating line is the locus of the points of maximum efficiency at various values of . The surge line represents the stable operation, the region to the right of the surge line being for stable conditions.

The performance of the compressor has been plotted in terms of volume flow rate and pressure ratio in Fig. 14.12.

The delivery pressure v's mass flow rate in a dynamic compressor is shown in Fig. 14.13. Suppose a discharge valve is put in the line for this compressor. The mass rate of flow will be zero when the discharge valve is closed and the static pressure developed is that delivered by the impeller to the air contained in the compressor. Such a situation is shown by point *a* in Fig. 14.13. If the valve is now opened, the flow of air takes place and the diffuser becomes effective in increasing the static pressure. This is shown by points *b* and *c*. The maximum delivery pressure is obtained at point *c*. As the valve is opened more, the mass flow rate increases beyond point *c* and the efficiency of compressor decreases

with decrease in delivery pressure. When the designed mass rate is greatly exceeded, the incidence between the vane and air angle becomes so large that flow separation and shock occur, accompanied by rapid decrease in efficiency.

If the compressor operates at some point *b* to the left of point *c*, then a decrease in mass rate of flow is accompanied by decrease in pressure developed by the compressor. If the static pressure of the air at compressor outlet does not decrease as rapidly as the developed pressure, there is a natural tendency of the air to flow back into the compressor in the direction of pressure gradient. With the drop in pressure at the

compressor outlet, the pressure gradient is reversed and also the direction of flow is reversed. Thus, this leads to unstable condition of cyclic reversal taking place at extremely high frequencies. This pulsating air flow phenomena is called 'surging'. Such a situation is avoided by keeping the operating point to the right of point *c,* because in this region a decrease in mass flow rate is accompanied by increase in delivery pressure, leading to stability.

Figure 14.13 *Surging and choking phenomena*

After point *c*, any increase in mass flow rate is accompanied by decrease in delivery pressure. This happens because the rate of increase in pressure loss due to the friction is more than the rate of

increase in pressure gain by diffuser. Theoretically, the decrease in delivery pressure is continued until the mass flow rate is represented by the point ‘*f*’. In practice, the maximum mass flow is limited by the point *e* because if the mass flow exceeds design mass flow, the air angles are widely different from vane angles and choking takes place. The point *e* represents the choking of the compressor, that is, the maximum mass flow rate condition.

Surging does not take place in the region *ce* as the reduction in mass flow rate is accompanied with the increase in delivery pressure and flow reversal is not possible and stability of operation is maintained.

Example 14.1

A centrifugal air compressor having a pressure ratio of 5, compresses air at a rate of 10 kg/s. If the initial pressure and temperature are 1 bar and 20°C respectively, find the final temperature of air and power required to drive the compressor. Take $\gamma = 1.4$ and $c_p = 1$ kJ/kg.K.

Solution

Given: $r_p = 5$, $\dot{m} = 10$ kg/s, $p_1 = 1$ bar, $T_1 = 273 + 20 = 293$ K, $\gamma = 1.4$, $c_p = 1$ kJ/kg.K

Power required to drive the compressor,

$P = \dot{m}c_p\,(T_2 - T_1) = 10 \times 1 \times (464 - 293) = 1710$ kW.

Example 14.2

A centrifugal compressor has a pressure ratio of 4:1 with an isentropic efficiency of 82% when running at 16,000 rpm. It takes in air at 17°C. Guide vanes at inlet give the air a pre whirl of 20° to the axial direction at all radii and the mean diameter of the eye is 200 mm, the absolute air velocity at inlet is 120 m/s. At exit, the blades are radially inclined and the impeller tip diameter is 550 mm. Calculate the slip factor of the compressor.

Figure 14.14 *T – s Diagram*

Solution

= 4, $T_1 = 273 + 17 = 290$ K, $N =$ 16,000 rpm, $\alpha_1 = 90° - 20° = 70°$

$d_1 = 200$ mm, $v_{a_1} = 120$ m/s, $\eta_{\text{isen}} =$ 0.82, $d_2 = 550$ mm, $\beta_2 = 90°$

Temperature after isentropic compression (Fig. 14.14)

Isentropic temperature rise,

$(\Delta T)_{\text{isen}} = 430.94 - 390 = 140.94$ K

Actual temperature rise,

Power input per unit mass flow rate

$= c_p \times (\Delta T)_a = 1.005 \times 171.88 = 172.74$ kJ/kg

An inlet to impeller (Fig. 14.15 (a)),

$v_{a1} = 120$ m/s

Angle of pre-whirl = 20°

At exit of impeller (Fig. 14.15(b))

Figure 14.15 *Velocity diagrams for centrifugal compressor: (a) Inlet, (b) Outlet*

For radial discharge, $v_{w2} = u_2 =$ 460.77 m/s

Power input per unit mass flow rate

$$= u_2 v'_{w2} - u_1 v_{w1}$$
$$= 460.77\ v'_{w2} - 167.55 \times 41.04 = 172.74 \times 10^3$$
$$v'_{w2} = 389.82 \text{ m/s}$$

Example 14.3

A centrifugal compressor running at 15000 rpm takes in air at 15°C and compresses it through a pressure ratio of 4 with an isentropic efficiency of 80%. The blades are radially inclined and the slip factor is 0.85. Guide vanes at inlet give the air an angle of pre-whirl of 20° to the axial direction. The mean diameter of the impeller eye is 200 mm and the absolute air velocity at inlet is 120 m/s. Calculate the impeller tip diameter. Take c_p = 1.005 kJ/kg.K and $\gamma = 1.4$.

Solution

Given: $N = 15000$ rpm, $T_1 = 273 + 15 = 288$ K, $r_p = 4$, $\eta_{\text{isen}} = 80\%$, $\phi_s = 0.85$, $v_{a1} = 120$ m/s, $d_1 = 200$ mm, $\alpha_1 = 70°$

Figure 14.16 *(a) T-s diagram, (b) Velocity triangles.*

From Fig. 14.16 (a), we have

Compressor work, $w_c = c_p \cdot \Delta T = 1.005 \times 175 = 175.875$ kJ/kg

From Fig. 14.16 (b), $v_{w1} = v_{a1} \cos 70°$

$$= 120 \cos 70° = 41.04 \text{ m/s}$$

For radial discharge, $\beta_2 = 90°$ and

$$v_{w2} = u_2$$

Power input = $u_2 - u_1\ v_{w1}$

$$175{\cdot}875 \times 10^3 = u_2 \times 0.85\ u_2 - 157 \times 41.04$$

or $u_2 = 463.13$ m/s

or $d_2 = 589.7$ mm

Example 14.4

The free air delivered by a centrifugal compressor is 25 kg/min. The suction condition is 1 bar and 25°C. The velocity of air at inlet is 50 m/s. The isentropic efficiency of the compressor is 70%. If the total head pressure ratio of the compressor is 4, determine (*a*) the total head temperature of air at the

exit of the compressor, and *(b)* brake power required to run the compressor assuming mechanical efficiency of 96%. Pressure and temperature of air at inlet are static. For air $\gamma = 1.4$ and $R = 0.287$ kJ/kg. K.

Solution

Figure 14.17 *T–s diagram for centrifugal compressor*

Refer to Fig. 14.17.

Brake power required to run the compressor =

Example 14.5

Air at a temperature of 20°C flows into the centrifugal compressor running at 21,000 rpm. The following data are given:

Slip factor = 0.85

Work input factor = 1.0

Isentropic efficiency = 72%

Outer diameter of blade tip = 60 cm

Assuming the absolute velocities of air entering and leaving the compressor are same, determine:

1. Temperature rise of air passing through compressor, and
2. Static pressure ratio

Take c_p = 1.005 kJ/kg. K.

Solution

Work done per kg of air = $u_2 v_{w2} - u_1 v_{w1}$

$$= (659.7)^2 \times 0.85 \times 10^{-3} \times 1 = 369.923 \text{ kJ/s}$$

or $T_2 = 293 + 0.72 \times 368.1 = 558$ K

Static pressure ratio,

Example 14.6

A centrifugal compressor handles 180 kg/min of air. The suction pressure and temperature are 1 bar and 300 K. The suction velocity is 90 m/s. The delivery conditions are 2 bar, 400 K, and 240 m/s. Calculate (a) the isentropic efficiency, (b) the power required to drive the

compressor, and (c) the overall efficiency of the unit. Take c_p = 1.005 kJ/kg. K. Assume that entire kinetic energy gained in the impeller is converted into pressure in the diffuser.

Solution

= 3 kg/s, p_1 = 1 bar, T_1 = 300 K, v_{a1} = 90 m/s, p_2 = 2 bar, T_2 = 350 K, v_{a2} = 240 m/s

1. Isentropic efficiency,

$$T_2 = 300 \times 1.219 = 365.7 \text{ K}$$

   Isentropic work done, $w_{\text{isen}} = c_p(T_2 - T_1) +$

   Actual work done in the impeller,

2. Power required to drive the compressor = $\dot{m} \times w_a = 3 \times 125.25 = 375.75$ kW
3. Let T_3 = temperature at the exit of diffuser

or $1.005\ (T_3 - 400) = 24.75$

or $T_3 = 424.6$ K

or $p_3 = 2 \times 1.2326 = 2.465$ bar

After isentropic compression, the delivery temperature from diffuser is,

Example 14.7

A centrifugal compressor handles 540 kg/min of air at 1 bar and 20°C ambient conditions. The air is compressed from 1 bar static pressure to 4.5 bar total pressure. The air enters the impeller eye with a velocity of 150 m/s with no pre-whirl. The ratio of whirl speed to tip speed is 0.9. Calculate (a) the rise in total temperature during

compression if the change in kinetic energy is negligible, (b) the tip diameter of the impeller, (c) the power required, and (d) the eye diameter if the hub diameter is 130 mm.

Assume that the compressor runs to 20,000 rpm with isentropic efficiency of 82%.

Solution

= 9 kg/s, $p_{01} = p_1 = 1$ bar, $T_1 = 273 + 20 = 293$ K, $N = 20{,}000$ rpm, $\eta_{isen} = 0.82$,

Figure 14.18 *T–s diagram*

$p_{02} = 4.5$ bar, $v_{a1} = 150$ m/s, = 0.9, $d_h = 130$ mm = 0.13 m

The T - s diagram is shown in Fig. 14.18.

1. Isentropic rise in total temperature,

$$(\Delta T)_{isen} = 450.3 - 304.2 = 146.13 \text{ K}$$

Actual rise in total temperature,

2. Actual work consumed by the compressor,

$$w_{actual} = c_p\,(\Delta T)_{actual} = 1.005 \times 178.2 = 179.1 \text{ kJ/kg}$$

Work consumed given by Euler's equation without pre-whirl,

Tip diameter of impeller,

3. Power required = $\dot{m} \times w_{actual} = 9 \times 179.1 = 1611.9$ kW

4. or $D_1 = 0.2848$ m or 284.8 mm

Example 14.8

A centrifugal compressor delivers

720 m^3/min of free air while running at 12,000 rpm. The ambient air conditions are 1 bar and 27°C. The air is compressed to 4.2 bar with an isentropic efficiency of 0.85. Blades are radial at the outlet of the impeller and the flow velocity is 60 m/s, which may be assumed constant throughout. The outer radius of impeller is twice the inner and the slip factor is 0.9. The blade area coefficient is 0.9 at inlet. Determine (a) the final temperature of air, (b) the theoretical power, (c) the inlet and outlet diameter of impeller, (d) the breadth of impeller at inlet, (e) the impeller blade angle at inlet, and (f) the diffuser blade angle at inlet.

Solution

p_1 = 1 bar, T_1 = 273 + 27 = 300 K, p_2 = 4.2 bar, η_{isen} = 0.85, v_{f2} = 60 m/s, $r_2 = 2r_1$, ϕ_s = 0.9, k_b = 0.9.

1.
2. Mass flow rate

   Theoretical power,

   = 13.94 × 1.005 (478.8 − 300) = 2504.4 kW

3. Work done =
4. Volume flow rate = $\pi D_1 b_1 k_b v_{f1}$
5. or β_1= 15.03°
6. or α_2 = 8.48°

Example 14.9

A centrifugal blower compresses

5.4 m^3/s of air from 1 bar and 25°C to 1.5 bar. The index of compression is 1.5. The flow velocity of 70 m/s is constant throughout. The inlet and outlet diameters of impeller are 0.30 m and 0.60 m, respectively. The blower rotates at 9000 rpm. Determine (a) the blade angles at inlet and outlet of the impeller, (b) the absolute angle at the tip of the impeller, and (c) the breadth of blade at inlet and outlet.

Assume that no diffuser is employed and the whole pressure increase occurs in the impeller and the blades have negligible thickness.

Solution

$\dot{V}_1 = 5.4$ m³/s, $p_1 = 1$ bar, $T_1 = 273 + 25 = 298$ K, $n = 1.5$, $p_2 = 1.5$ bar, $v_{f_1} = v_{f_2} = 70$ m/s, $D_1 = 0.3$ m, $D_2 = 0.6$ m, $N = 9000$ rpm

$$u_2 = 2u_1 = 282.74 \text{ m/s}$$

1.

or $\beta_1 = 26.34°$

or $\beta_2 = 28.38°$

2. or $\alpha_2 = 24.56°$

3. $\dot{V}_1 = \pi D_1 b_1 v_{f1}$

Example 14.10

A centrifugal compressor delivers 18 kg/s of air with a total head pressure ratio of 4 : 1. The speed of the compressor is 16000 rpm. Total head temperature at inlet is 22°C,

slip factor 0.9, power input factor 1.05 and isentropic efficiency 0.82. Calculate (a) the overall diameter of the impeller and (b) the power input.

Solution

$\dot{m} = 18$ kg/s, $r_{op} = 4$, $N = 16000$ rpm, $T_{01} = 273 + 22 = 295$ K, $\phi_s = 0.9$, $\phi_w = 1.05$, $\eta_{\text{isen}} = 0.82$

1. or $u_2 = 431.2$ m/s

2. Power input, $P = \dot{m}\, c_p\, (T_{02}' - T_{01})$

$= 18 \times 1.005\ (469.8 - 295) = 3162.1$ kW

Example 14.11

A single-sided centrifugal compressor impeller fitted with a diffuser, without water cooling, requires 2000 kW when running at 16,500 rpm. The outlet diameter of impeller is 0.5 m and the width of the casing of the vortex chamber between the impeller and the diffuser is 50 mm. During a test on this compressor, static temperature and pressure were measured at a radius of 0.28 m and were found to be 120°C and 3 bar, respectively. Assuming whirl speed at the tip of the impeller to be 0.95 of the peripheral speed and neglecting friction in vortex chamber, calculate (a) the flow rate, (b) the resultant speed at the section given, and (c)

the total temperature at the section given. Take $c_p = 1.005$ kJ/kg, K, $\gamma = 1.4$ and intake pressure = 1 bar.

Solution

$P = 2000$ kW, $N = 16500$ rpm, $D_2 = 0.5$ m, $b = 50$ mm, $\phi_s = 0.95$, $p_1 = 1$ bar

1. Work done = $v_{w2}u_2$
2. Using suffix 3 for the given section

   Area of flow, $A_{f3} = 2\pi r_3 b_3 = 2\pi \times 0.28 \times 50 \times 10^{-3} = 0.088$ m^2

   Volume rate of flow at the given section,

   For free vortex in vaneless space, $v_{w1} \times r_1 =$ const.

   i.e., $v_{w1}r_1 = v_{w2}r_2 = v_{w3}r_3$

3.

Example 14.12

A centrifugal compressor delivers 580 m^3/min of free air when running at 8000 rpm. Use the following data: inlet pressure and temperature of air = 1.013 bar and 20°C, pressure ratio = 3.5, isentropic efficiency = 83 %, flow velocity throughout the compressor = 62 m/s, the blades are radial at the outlet of the impelier, tip diameter = 2 times eye diameter, blade area coefficient = 0.94. Find (a) the input power required to run the compressor, (b) the impeller diameters at inlet and outlet, (c) the breadth of impeller at inlet, and (d) the impeller blade angle at inlet.

Solution

Given that $\dot{V}_a$ = 580 m^3/min, N = 800 rpm, p_1 = 1.013 bar, T_1 = 273 + 20 = 293 K, r_p = 3.5, η_{isen} = 0.83, v_f = 62 m/s, β_2 = 90°, $D_2 = 2D_1$, k_t = 0.94

Mass flow rate of air,

1. Input power required

   = 1777.72 kW
2. For radial blades, work done on compressor,
3. $\dot{V}_a = \pi D_1 b_1 v_f k_t$

   or = $\pi \times 0.4665 \times b_1 \times 62 \times 0.94$

   or b = 0.113 m or 113 mm
4. $\beta_1 = 17.6°$

Example 14.13

A centrifugal compressor runs at 16,000 rpm and 2300 kW of power is required to run the compressor. The outer diameter of the impeller is 50 cm. The uniform width of the casing of vortex chamber between impeller and diffuser is 4 cm. Static conditions at a radius of 27 cm were measured and it were equal to 2.4 bar and 390 K. Surrounding pressure is 1 bar. Assuming slip factor of 0.94, find (a) the mass flow rate of gas, (b) the resultant velocity and (c) the temperature at the given radius of 27 cm. Assume $c_p = 1.005$ kJ/kgK and $\gamma = 1.4$.

Solution

Given that $N = 16{,}000$ rpm, $P = 2300$ kW, $D_2 = 50$ cm, $B = 4$ cm, At $R = 27$ cm, $p = 2.4$ bar, $T = 390$ K, $p_1 = 1$ bar, $\phi_s = 0.94$, $c_p = 1.005$ kJ/kg.K, $\gamma = 1.4$

1.
2. Using suffix 3 for the given section,

   Area of flow, $A_{f3} = 2\pi R_3 B_3 = 2\pi \times 0.27 \times 0.04 = 0.06786 \text{ m}^2$

   Volume rate of flow,

   For free vortex in vaneless space, $v_{w1} \times r_1 =$ const.

   i.e., $v_{w1}r_1 = v_{w2}r_2 = v_{w3}r_3$

   1. $= 460.98 \text{ K} = 461 \text{ K or } 188°\text{C}$

Example 14.14

A centrifugal compressor has to deliver 6kg/s of air with pressure

ratio of 4:1 at 17500 rpm. Initial conditions are static air at 1 bar and 15°C. Assuming an adiabatic efficiency of 78%, ratio of whirl speed to tip speed 0.94 and neglecting all other losses, calculate the tip speed, diameter, and rise in total pressure. Determine the external diameter of eye for which the internal diameter is 12 cm and the axial velocity at inlet is 150 m/sec. Assume c_p = 1.005 kJ/kgK and γ = 1.4.

Solution

Given that: $\dot{m}_a$ = 6 kg/s, = 4, N = 17500 rpm, p_{01}= 1 bar, T_{01} = 273 + 15 = 288 K, η_{isen} = 0.78, = 0.94, d_h = 12 cm, for radial inlet, α_1 = 90°

and $v_{f1} = v_{a1} = 150$ m/s, $c_p = 1.005$ kJ/kg.K, $\gamma = 1.4$

The *T-s* diagram is shown in Fig. 14.19(a) and the impeller is shown in Fig. 14.19(b)

Figure 14.19 *Diagrams for centrifugal compressor: (a) T-s diagram, (b) Impeller*

Let $v_{a1} = v_{a2}$, then

Rise in total pressure, $\Delta p_0 = p_{02} - p_{01} = 3.82 - 1 = 2.82$ bar

Work required per kg of air, $w = c_p(T_{02}' - T_{01})$

$$= 1.005(460.4 - 288) = 173.262 \text{ kJ/kg}$$

Also, work required per kg of air,

or $d_2 = 0.468$ m or 46.8 cm

Example 14.15

A single-sided centrifugal air compressor delivers 1800 kg of air per minute. The air enters the eye of the impeller axially at total pressure of 100 kPa and total temperature of 290 K. The overall diameter of the impeller is 700 mm and it rotates at 1600 rpm. The slip factor is 0.9 and the work input factor is 1.1. The isentropic efficiency is 85%. Calculate (a) power required by the

compressor, (b) pressure coefficient, and (c) total pressure at delivery.

[IES, 2003]

Solution

Given that $\dot{m}_a = 1800$ kg/min, $\alpha_1 = 90$, $p_{01} = 100$ kPa, $T_{01} = 290$ K, $D_2 = 700$ mm, $N = 1600$ rpm, $\phi_s = 0.9$, $\phi_w = 1.1$, $\eta_{isen} = 85\%$

1.

2. Pressure efficient, $\phi_p = \phi_w\ \phi_s \eta_{isen} = 1.1 \times 0.9 \times 0.85 = 0.8415$
3.
   or $p_{02} = 100 \times 1.00282 = 100.82$ kPa

Example 14.16

A centrifugal compressor running at 18,000 rev/min takes in air at 25° C and compresses it through a

pressure ratio of 4.0 with an isentropic efficiency of 80%. The guide vanes at inlet give the air an angle of pre-whirl of 20° to the axial direction. The mean diameter of impeller eye is 225 mm. The absolute air velocity at inlet is 130 m/s. At exit the blades are radially inclined. If the slip factor is 0.90, calculate the impeller tip diameter.

[IAS, 1999]

Solution

Given that $N = 18000$ rpm, $T_1 = 25 + 273 = 298$ K, $r_p = 4$, $\eta_{\text{isen}} = 80\%$, pre-whirl at inlet = 20° or $\alpha_1 = 90° - 20° = 70°$, $d_m = 225$ mm, $v_{a1} = 130$ m/s, $\beta_2 = 90°$, $\phi_s = 0.90$

Temperature after isentropic compression,

Isentropic temperature rise,

$$(\Delta T)_{\text{isen}} = T_2 - T_1 = 442.83 - 298 = 144.83 \text{ K}$$

Actual temperature rise,

Power input per unit mass flow rate $= c_p \times (\Delta T)_{\text{act}} = 1.005 \times 181 = 181.9$ kJ/kg

The velocity diagrams are shown in Fig. 14.20

$v_{w1} = v_{a1} \cos \alpha_1 = 130 \cos 70° = 44.46$ m/s

For radial discharge, $v_{w2} = u_2$

Slip factor,

$= 0.9\ v_{w2} = 0.9 \times 0.9425\ d_2 = 0.84825 \times d_2$ m/s

Figure 14.20 *Velocity diagrams: (a) Inlet, (b) Outlet*

Power input per unit mass flow rate $= u_2 - u_1\ v_{w1}$

or $181.9 \times 10^3 = 0.9425 \times d_2 \times 0.84825 \times d_2 - 212 \times 44.46$

$$= 0.7995 - 9425.52$$

or $d_2 = 488$ mm

Example 14.17

A double-sided centrifugal compressor has root and tip diameters of 18 cm and 30 cm and is to deliver 16 kg of air per second at 16,000 rpm. The design ambient conditions are 15°C and 1 bar and the compressor has to be a part of a stationary power plant. Determine the following:

1. Suitable values for impeller vane angles at the root and tip of the eye if the air is given 20° of pre-whirl at all radii. The axial component of the velocity is constant throughout the impeller and is 150 m/s.
2. Power required if the power input factor is 1.05 and mechanical efficiency is 95%
3. Maximum Mach number at the eye

Take for air: c_p = 1.005 kJ/kg K and γ = 1.4.

[IAS, 1998]

Solution

Given that r_1 = 9 cm, r_2 = 15cm, $\dot{m}_a$ = 16 kg/s, N = 16000 rpm, T_1 = 273

$+ 15 = 288$ K

$p_1 = 1$ bar, $\alpha_1 = \alpha_2 = 70°$, $v_{f1} = v_{f2} = 150$ m/s, $\phi_s = 1.05$, $\eta_{mech} = 95\%$, $c_p = 1.005$ kJ/kg K, $\gamma = 1.4$

1. Angular speed of rotor,

The velocity diagrams are shown in Fig. 14.21.

$u_1 = \omega r_1 = 1675.5 \times 0.09 = 150.8$ m/s

$u_2 = \omega r_2 = 1675.5 \times 0.15 = 251.3$ m/s

$\alpha_1 = 90°$ - angle of pre-whirl $= 70°$

Figure 14.21 *Velocity diagrams*

Work done per kg of air, $w = v_{w2}\, u_2 - v_{w1}\, u_1 = v_{w1}\,(u_1 - u_1)$

$$= 54.6\ (251.3 - 150.8) = 5487.3$$

2. Power required
3.

Summary for Quick Revision

1. A centrifugal compressor is of rotodynamic type in which the pressure rise is primarily due to the centrifugal action.
2. Stagnation or total head values:

1. Stagnation temperature,
2. Stagnation enthalpy,
3. Stagnation pressure,

3. Isentropic efficiency,
4. Ideal velocity diagrams:
    1. $\alpha_1 = 90°$, $\beta_2 = 90°$, $v_{a1} = v_{f1}$ and $v_{r2} = v_{f2}$, i.e. $v_{w1} = 0$ and $v_{w2} = u_2$
    2. Torque, $T = u_2 r_2$
    3. Work done for 1 kg/s of air flow,
    4. From SFEE, $w = c_p (T_{02} - T_{01}) = h_{02} - h_{01}$

       $= c_p (T_2 - T_1) = h_2 - h_1$ for $v_{a1} = v_{a2}$

    5.
    6. Stagnation pressure ratio:
    7.
    8.
    9. Total temperature increase across an impeller,
    10. Indicated power,
5. Actual velocity diagrams:
    1. Work done on 1 kg/s of air, $w =$
    2. ΔKE
6. Width of blades:

   where k_b = blades factor, b = width of height of blades, t = thickness of blades,

   v_s = specific volume

7. Slip factor,

   Slip $= u_2 - v_{w2}$

8. Pressure coefficient,

9. Work factor,
10. Relationship between ϕ_p, ϕ_w, ϕ_s and η_{isen}:

$$\phi_p = \phi_s \, \phi_w \, \eta_{\text{isen}}$$

11. Shape of impeller blades:
    1. There are three types of impeller blades:
12. Backward curved blades, $\beta_2 < 90°$.

    1. a. Radial blades, $\beta_2 = 90°$.
    2. b. Forward curved blades, $\beta_2 > 90°$.

    Slightly backward curved impeller blades give optimum efficiency. Its consequences are:

    3. a. Degree of reaction increases which results in increasing part of kinetic energy transferred into pressure energy within the impeller.
    4. b. The length of impeller channel increases which increases the friction loss.
    5. c. Radial blades are used for aircraft centrifugal compressor.

13. The diffuser converts the kinetic energy imparted to air by the impeller into pressure rise.
14. Pre-whirl is given to intake guide vanes in order to avoid acceleration to sonic or supersonic velocities to keep the Mach number below 0.9 of the air flow entering the impeller eye. This increases the work done on the air $v_{w1} \neq 0$.
15. Performance characteristics:

For D = const

where G = mass rate of flow, η = rotor rpm, p_1 = inlet pressure,

T_1 = inlet temperature, p_3 = pressure at diffuser outlet.

16. The pulsating air flow phenomena is called surging.

    The maximum mass flow rate of air is called choking of the compressor.

Multiple-choice Questions

1. The inlet and exit velocity diagrams of a turbomachine rotor are shown in Fig. 14.22.

   This turbomachine is

   1. An axial compressor with backward curved blades
   2. A radial compressor with backward curved blades
   3. A radial compressor with forward curved blades
   4. An axial compressor with forward curved blades

   Figure 14.22 *Velocity triangles*

2. It is recommended that the diffuser angle should be kept less than 18° because
   1. Pressure decreases in flow direction and flow separation may occur
   2. Pressure decreases in flow direction and flow may become turbulent
   3. Pressure increases in flow direction and flow separation may occur
   4. Pressure increases in flow direction and flow may become turbulent
3. When the outlet angle from the rotor of a centrifugal compressor is more than 90, then the blades are said to be
   1. Forward curved
   2. Backward curved
   3. Radial
   4. Either backward or forward curved
4. The degree of reaction of a turbomachine is defined as the ratio of the
   1. Static pressure change in the rotor to that in the stator
   2. Static pressure change in the rotor to that in the stage
   3. Static pressure change in the stator to that in the rotor
   4. Total pressure change in the rotor to that in the stage

5. If two geometrically similar impellers of a centrifugal compressor are operated at the same speed, their head, discharge, and power will vary with their diameter ratio d as
   1. d, d^2, and d^3, respectively
   2. d^2, d^3, and d^5, respectively
   3. d, d^3, and d^5, respectively
   4. d^2, d, and d^3, respectively
6. Which one of the following velocity triangles (Fig. 14.23) represents the one at the exit of a radial impeller with forward curved blades?

   (u_2 = *peripheral velocity*, v_2 = *absolute velocity*, w_2 = *relative velocity*)

Figure 14.23 *Velocity triangles*

7. The stagnation pressure rise in a centrifugal compress or stage takes place
   1. Only in the diffuser
   2. In the diffuser and impeller
   3. Only in the impeller
   4. Only in the inlet guide vanes
8. For 15 m^3/s air flow at 10 mm Hg head, which one of the following would be the best choice?
   1. Centrifugal fan with forward curved blades
   2. Axial fan with a large number of blades in rotor
   3. Axial propeller fan with a few blades in rotor
   4. Cross-flow fan
9. What is the ratio of the isentropic work to Euler's work known as?
   1. Pressure coefficient
   2. Slip factor
   3. Work factor
   4. Degree of fraction
10. Which of the following is the effect of blade shape on performance of centrifugal compressor?
    1. Backward curved blade has poor efficiency
    2. Forward curved blades have higher efficiency
    3. Backward curved blades lead to stable performance
    4. Forward curved blades produce lower pressure ratio
11. Surging basically implies
    1. Unsteady, periodic, and reversed flow
    2. Forward motion of air at a speed above sonic velocity

3. The surging action due to the blast of air produced in a compressor
4. Forward movement of aircraft

12. Centrifugal compressors are suitable for large discharge and wider mass flow range, but at a relatively low discharge pressure of the order of 10 bars, because of
    1. Low pressure ratio
    2. Limitation of size of receiver
    3. Large speed
    4. High compression index
13. Given: v_{w2}= velocity of whirl at outlet

    u_2 = peripheral velocity of the blade tips.

    The degree of reaction in a centrifugal compressor is equal to

    1.
    2.
    3.
    4.
14. In a centrifugal compressor assuming the same overall dimensions, blade inlet angle, and rotational speeds, which of the following blades will give the maximum pressure rise?
    1. Forward curved blades
    2. Backward curved blades
    3. Radial blades
    4. All three types of bladings have the same pressure rise
15. Under which one of the following sets of conditions will a surpersonic compressor have the highest efficiency?
    1. Rotor inlet velocity is supersonic and exit velocity subsonic; stator inlet velocity is subsonic and exit velocity is subsonic
    2. Rotor inlet velocity is supersonic and exit velocity subsonic; stator inlet velocity is supersonic and exit velocity is subsonic
    3. Rotor inlet velocity is supersonic and exit velocity supersonic; stator inlet is supersonic and exit velocity is subsonic
    4. Rotor inlet velocity is supersonic and exit velocity supersonic; stator inlet velocity is subsonic and exit

velocity is subsonic

16. The curve in Fig. 14.24 shows the variation of theoretical pressure ratio with mass of flow rate for a compressor running at a constant speed. The permissible operating range of the compressor is represented by the part of the curve from
    1. A to B
    2. B to C
    3. B to D
    4. D to E

Figure 14.24

17. In a centrifugal compressor, the highest Mach number leading to shockwave in the fluid flow occurs at
    1. Diffuser inlet radius
    2. Diffuser outlet radius
    3. Impeller inlet radius
    4. Impeller outer radius
18. In the centrifugal air compressor design practice, the value of polytropic exponent of compression is generally taken as
    1. 1.2
    2. 1.3
    3. 1.4
    4. 1.5
19. For centrifugal compressors, which one of the following is the correct relationship between pressure coefficient (ϕ_p), slip factor (ϕ_s), work input factor (ϕ_n) and isentropic efficiency (η_a)?
    1.
    2.
    3.
    4.
20. At the eye tip of a centrifugal impeller, blade velocity is 200 m/s while the uniform axial velocity at the inlet is 150m/s. If the sonic velocity is 300 m/s, the inlet Mach number of the flow will be
    1. 0.50
    2. 0.66
    3. 0.83
    4. 0.87
21. What will be the shape of the velocity triangle at the exit of a radial bladed centrifugal impeller, taking into account slip?
    1. Right-angled
    2. Isosceles

3. All angles less than 90°
4. One angle greater than 90°

22. What does application of centrifugal air compressors lead to?
    1. Large frontal area of aircraft
    2. Higher flow rate through the engine
    3. Higher aircraft speed
    4. Lower frontal area of the aircraft
23. Consider the following statements:

    In centrifugal compressors, there is a tendency of increasing surge when

    1. the number of diffuser vanes is less than the number of impeller vanes
    2. the number of diffuser vanes is greater than the number of impeller vanes
    3. the number of diffuser vanes is equal to the number of impeller vanes
    4. mass flow is greatly in excess of that corresponding to the design mass flow

    Which of these statements is/are correct?

24. In centrifugal compressor terminology, vaneless space refers to the space between
    1. The inlet and blade inlet edge
    2. Blades in the impeller
    3. Diffuser exit and volute casing
    4. Impeller tip and diffuser inlet edge
25. Match List-I with List-II (pertaining to blower performance) and select the correct answer using the codes given below the list:

    Codes:

	A	B	C
1.	4	3	2

2. 1 3 2
3. 4 1 3
4. 2 3 4

26. The flow in the vaneless space between the impeller exit and diffuser inlet of a centrifugal compressor can be assumed as
    1. Free vortex
    2. Forced vortex
    3. Solid body rotation
    4. Logarithmic spiral
27. Which portion of the centrifugal compressor characteristics shown in Fig. 14.25 is difficult to obtain experimentally?

Figure 14.25

    1. RS
    2. ST
    3. TU
    4. UV
28. The pressure rise in the impeller of centrifugal compressor is achieved by
    1. The decrease in volume and diffusion action
    2. The centrifugal action and decrease in volume
    3. The centrifugal and diffusion action
    4. The centrifugal and push-pull action
29. In a radial blade centrifugal compressor, the velocity of blade tip is 400 m/s and slip factor is 0.9. Assuming the absolute velocity at inlet to be axial, what is the work done per kg of flow?
    1. 36 kJ
    2. 72 kJ
    3. 144 kJ
    4. 360 kJ
30. The power required to drive a turbo-compressor for a given pressure ratio decreases when
    1. Air is heated at entry
    2. Air is cooled at entry
    3. Air is cooled at exit
    4. Air is heated at exit
31. In a centrifugal compressor, how can the pressure ratio be increased?
    1. Only by increasing the tip speed
    2. Only by decreasing the inlet temperature
    3. By both (a) and (b)
    4. Only by increasing the inlet temperature

Explanatory notes

1. 20. (a)
2. 31. (c)

Work done

Review Questions

1. What are the constructional features of a centrifugal compressor?
2. Explain the principle of working of a centrifugal compressor.
3. How does the velocity and pressure of air vary through the impeller and the diffuser?
4. What are the various types of impellers?
5. Define total head values.
6. Define isentropic efficiency.
7. Define stagnation pressure ratio.
8. Define slip factor.
9. What is work factor and pressure coefficient?
10. What is the relationship between ϕ_p, ϕ_w, ϕ_s, and ϕ_{isen}?
11. List the losses which generally occur in a centrifugal compressor.
12. What are the effects of impeller blade shapes on performance?
13. What is the role of a diffuser?
14. Explain pre-whirl and its role.
15. Explain the phenomena of surging and choking.

Exercises

14.1 A single eye, single-stage centrifugal compressor delivers 20 kg/s of air with a pressure ratio of 4 when running at 15,000 rpm. The pressure and

temperature of air at the inlet are 1 bar and 17°C. Assume the following:

Slip factor = 0.9, work input factor = 1.04, isentropic efficiency = 75%.

Calculate (a) the IP required to drive the compressor and (b) the blade angles at the impeller eye if the root and tip diameters are 20 cm and 40 cm, respectively.

14.2 A centrifugal compressor delivers 600 m^3/min of free air when running at 900 rpm. The following data is given:

Inlet pressure and temperature of air = 1 bar and 20°C

Compression ratio = 3.5

Isentropic efficiency = 83%

Flow velocity throughout the impeller = 60 m/s

Tip diameter = 2 × eye diameter

Blade area coefficient = 0.94

The blades are radial at outlet of impeller.

Calculate (a) the IP required to run the compressor, (b) impeller diameters at inlet and outlet, (c) breadth of impeller at inlet, and (d) impeller blade angle at inlet.

14.3 The first stage of a centrifugal compressor is single-inlet with an eye

60 cm, hub 20 cm and an overall diameter of 100 cm. When supplied with 20°C air at 1 bar and running at 6000 rpm, it compresses 30 kg/s of air through a compression ratio of 2.0.

Calculate (a) the isentropic efficiency of rotor based on impeller tip velocity with slip factor as 0.9, and (b) the rotor power and shaft power. Take η_{mech} = 0.97.

14.4 A single-sided centrifugal compressor is to deliver 14 kg/s of air when operating at a pressure ratio of 4:1 and a speed of 12000 rpm. The total head inlet conditions may be taken as 288 K and 1 bar. Assuming a slip factor of 0.9, a power input factor of 1.04 and an isentropic efficiency (based on the

total head) of 80%, estimate the overall diameter of the impeller. If the Mach number is not to exceed unity at the impeller tip and 50% of the losses are assumed to occur in the impeller, find the minimum possible depth of the diffuser.

14.5 The following data refer to a single-sided centrifugal compressor:

Air mass flow rate = 9 kg/s, Eye tip diameter = 0.3 m, Slip factor = 0.9, Isentropic efficiency = 80%.

Eye root diameter = 0.15 m, overall diameter of impeller = 0.5 m, Power input factor = 1.04, Rotational speed = 18000 rpm, Inlet stagnation pressure and temperature = 1.1 bar, 295 K.

Calculate (a) pressure ratio of the compressor, (b) inlet angle of the impeller vane of the root and tip radii of the eye, and (c) the axial depth of the impeller channel at the periphery of the impeller.

14.6 An air-craft fitted with a single-sided centrifugal compressor flies with a speed of 850 km/h at an altitude where the pressure is 0.23 bar and the temperature is 217 K. The inlet duct of the impeller eye contains fixed vanes which give the air pre-whirl of 25° at all radii. These inner and outer diameters of the eye are 180 mm and 330 mm, respectively, the diameter of the impeller tip is 540 mm, and the rotational speed is 16,000 rpm. Estimate

the stagnation pressure at the compressor outlet when the mass flow is 216 kg/min. Neglect losses in inlet duct and fixed vanes, and assume that the isentropic efficiency of the compressor is 0.80. Take the slip factor as 0.9 and the power input factor as 1.04.

14.7 A centrifugal compressor compresses 30 kg/s of air at a rotational speed of 1500 rpm. The air enters the compressor axially, and the conditions at the exit section are radius = 0.3 m, relative velocity of air at the tip = 100 m/s at an angle of 80°. Find the torque and power required to drive the compressor and work done.

14.8 Air at a temperature of 300 K flows in a centrifugal compressor running at

18,000 rpm. Slip factor = 0.8; isentropic total head efficiency = 0.75; outer diameter of blade tip = 0.5 m. Determine (a) the temperature rise of air and (b) static pressure ratio.

Assume that the velocities of air at inlet and exit of the compressor are the same.

14.9 A single inlet type centrifugal compressor delivers 8 kg/s of air. The ambient air conditions are 1 bar and 20°C. The compressor runs at 22,000 rpm with isentropic efficiency of 82%. The air is compressed in the compressor from 1 bar static pressure to 4.2 bar total head pressure. The air enters the impeller eye with a velocity of 150 m/s with no pre-whirl. Assuming that the ratio of whirl speed to tip speed is 0.9,

calculate (a) the rise in total head temperature during compression if the change in kinetic energy is negligible, (b) the tip diameter of the impeller, (c) power required, and (d) the eye diameter if the hub diameter is 10 cm.

14.10 A centrifugal compressor is to have a pressure ratio of 3.5:1. The inlet eye of the impeller is 30 cm in diameter. The axial velocity at inlet is 130 m/s and the mass flow rate is 10 kg/s. The velocity in the delivery duct is 115 m/s. The tip speed of the impeller is 450 m/s and runs at 16,000 rpm with total head isentropic efficiency of 78% and pressure coefficient of 0.72. The ambient conditions are 1 bar and 15°C. Calculate: (a) the static pressure ratio,

(b) the static pressure and temperature at inlet and outlet of compressor, (c) work of compressor per kg of air, and (d) the theoretical power required.

14.11 The following details refer to a centrifugal air compressor of a total head pressure ratio of 3 and isentropic efficiency of 70%. Find the power required to run the compressor.

Suction conditions:

Pressure = 1 bar,

Temperature = 20°C

Velocity of air at inlet = 60 m/s

Pressure and temperature are static

values.

Free air delivered = 20 kg/min

Mechanical efficiency = 95%

Ratio of specific heats for air = 1.4

Specific gas constant for air = 0.287 kJ/kg.K

14.12 A centrifugal compressor has a pressure ratio of 4:1 with an isentropic efficiency of 82% when running at 16000 rpm. It takes in air at 17°C. Guide vanes at inlet give the air a pre-whirl of 20° to the axial direction at all radii and the mean diameter of the eye is 200 mm; the absolute air velocity at inlet is 120 m/s. At exit, the blades are

radially inclined and the impeller tip diameter is 550 mm. Calculate the slip factor of the compressor.

ANSWERS TO MULTIPLE-CHOICE QUESTIONS

1. a
2. c
3. a
4. b
5. d
6. b
7. b
8. a
9. a
10. c
11. a
12. a
13. a
14. a
15. c
16. c
17. b
18. c
19. c
20. a
21. c
22. a
23. d
24. d
25. b
26. b
27. a
28. b
29. c
30. b
31. c

Chapter 15

Axial Flow Air Compressors

15.1 ❐ INTRODUCTION

In an axial flow compressor, the air flows throughout the compressor parallel to its axis. Axial flow compressor has many merits over centrifugal compressor and is particularly suitable for super charging of internal combustion engines, turbojets, and gas turbines.

15.2 ❐ CONSTRUCTIONAL FEATURES

The cross-section of a typical axial flow compressor is shown in Fig. 15.1. It consists of adjacent rows of moving and

fixed blades. The moving blades are mounted on the rotating drum and the fixed blades are fixed to the casing or stator. One stage of the compressor comprises a row of moving blades, followed by a row of fixed blades.

The blades belong to the aerofoil section designed on the basis of aerodynamic theory. Such a design prevents losses due to shock and turbulence and is free from stalling problems. The blades are said to be stalled when the air stream fails to follow the blade profile.

Figure 15.1 *Typical axial flow compressor*

The blades of a reaction gas turbine have profile formed by a number of circular arcs. This occurs because the acceleration process carried out in the

converging blade passages of a reaction turbine is much more efficient and stable as against the diffusing or decelerating process carried out in the diverging passage between the blades of an axial flow compressor.

To keep the flow velocity constant throughout the compressor length, the annular area is reduced from inlet to outlet of the compressor. In the diverging passage of the moving blades, there is a rise in temperature due to diffusion and the absolute velocity is also increased due to work input.

The fixed blades serve to convert a part of the kinetic energy of air into pressure energy by diffusion and also guide and redirect the air stream to enter the next

stage without shock.

15.3 ❐ WORKING PRINCIPLE

The axial compressor is generally driven by an internal combustion engine or a turbine. The work input to the rotor shaft is transferred by the rotor moving blades to the air, thus accelerating it. The space between the moving blades form diffuser passages so that the velocity of air relative to the blades is decreased as the air passes through them and consequently, the pressure rises. The air is further diffused in the fixed blades which also form diffuser passages. In the fixed blades, the air is turned through an angle so that its direction is such that it can pass to the next ring of moving blades without

shock. Generally, 5 to 14 stages are used.

15.4 ❐ SIMPLE THEORY OF AEROFOIL BLADING

When an object is placed within a fluid stream, a force is exerted on the object which may be inclined to the flow direction. The component of this force parallel to the direction of flow is called the drag and the normal component is called the lift, as shown in Fig. 15.2.

Consider an aerofoil section as shown in Fig. 15.3, which has an angle of attack α, the chord length C, and the length of the foil perpendicular to the plane of the paper, termed as the span, l. The resultant force F executed by the fluid on the aerofoil has two mutually perpendicular components: the lift F_L

and the drag F_D. Consider a surface element of elementary area dA of aerofoil on which pressure p acts. Let the tangent to dA make an angle θ with the flow direction. The differential lift and drag on this area element are as follows:

$$dF_L = pdA \cos \theta$$
$$dF_D = pdA \sin \theta$$

Figure 15.2 *Definition sketch of lift and drag*

Figure 15.3 *Pressure distribution on an aerofoil section*

Lift, $F_L = \int_A pdA\cos\theta$

Drag, $F_D = \int_A pdA\sin\theta$

where $\int_A$ represents the integration over the entire body surface area A.

We define the following lift and drag coefficients:

where ρ = density of air stream

U_∞ = free stream velocity of air

A = projected area of the body perpendicular to the oncoming flow, i.e., the frontal area of the body.

Let

$\dot{V}$ = free air delivered by the compressor, m^3/min

i = number of moving blades per blade ring

p_1, T_1 = ambient air conditions

D_m = blade ring mean diameter, m

k_b = blade factor

h = blade height, m

N = rotor speed, rpm

C_L, C_D = lift and drag coefficients respectively

A = projected area of blades, m^2

Then density of air,

$$\dot{V} = v_a A$$
$$A = k_b \pi D_m h$$

Figure 15.4 *Velocity triangle*

From the velocity triangle as shown in Fig. 15.4, we get

Power input/stage,

where and η_c = efficiency of the compressor

15.5 ❐ VELOCITY DIAGRAMS

Figure 15.5 shows the velocity triangles for one stage of an axial flow compressor. All angles are measured from the axial direction. The blade velocity u is taken to be same at blade entry and exit as the air enters and leaves the blades at almost equal radii. Due to the diffusion process in the moving blades, $v_{r2} < v_{r1}$ but $v_{a2} > v_{a1}$. The air leaves the fixed blades with velocity v_{a3} at an angle α_3 and is redirected to the next page. It is assumed that $v_{a1} = v_{a3}$.

From the inlet velocity triangle, we have

Figure 15.5 *Velocity diagrams for axial flow compressor*

From the outlet velocity triangle, we have

Assuming 1 kg flow of air through the compressor stage, we have

Tangential force per kg of air = $v_{w2} - v_{w1}$

Work required by the stage per kg of air,

Now $v_{f1} = v_{f2} = v_f$

Also, isentropic efficiency of the stage,

15.6 ❐ DEGREE OF REACTION

The degree of reaction, R_d, is defined as the ratio of pressure rise in the rotor blades to the pressure rise in the compressor stage.

Pressure rise in the rotor blades

Pressure rise in the compressor stage

For 50% degree of reaction,

From Eqs (15.8) and (15.9), we have

Therefore, with 50% degree of reaction, the blades are symmetrical. This reduces

the friction losses considerably.

15.7 ❒ PRESSURE RISE IN ISENTROPIC FLOW THROUGH A CASCADE

Consider the incompressible isentropic and steady flow through a cascade from condition 1 to condition 2. Using Bernoulli's equation, we have

Now, $v_{f1} = v_{f2} = v_f$

15.8 ❒ POLYTROPIC EFFICIENCY

It is the isentropic efficiency of one stage of a multistage compressor. This small-stage efficiency is constant for all stages of a compressor with infinite number of stages.

Consider the compression process of a multistage compressor on a T-s diagram as shown in Fig. 15.6 from total pressure p_{01} to p_{02} in four stages of equal pressure ratio with intermediate pressure p_{0a}, p_{0b}, and p_{0c}.

Overall isentropic stagnation efficiency of machine is,

Stagnation isentropic efficiency for the stage,

Total actual temperature rise,

Figure 15.6 *Concept of polytropic efficiency*

Now $(\Delta T_0)_{m/c} = (1 - a) + (a - b) + (b - c) + (c - 2)$

$$\Sigma\,(dT_0)_{st} = (1 - a) + (a' - b'') + (b' - c'') + (c' - 2'')$$

On the T-s plot, the constant pressure lines diverge towards the right, therefore,

$$(a' - b'') > (a - b)$$
$$(b' - c'') > (b - c)$$

and so on.

Therefore, we can say that

$$\Sigma\,(dT_0)_{st} > (\Delta T_0)_{m/c}$$

$$\therefore\ \eta_{\text{isen (st)}} > \eta_{\text{isen }(m/c)}$$

The small stage efficiency $\eta_{\text{isen (st)}}$ which is constant for all stages is called polytropic efficiency and is denoted by η_p.

Let the law of compression for the irreversible adiabatic path 1 – 2′ be,

and for the isentropic path 1–2 is,

Eq. (15.20) can be written as:

Differentiating, we get

Differentiating, we get

Eq. (15.21) becomes

Similarly, for the ideal compression process we have

From Eq. (15.23), we have

Integrating between the end states 1 and 2′, we get

15.9 ❐ FLOW COEFFICIENT, HEAD OR WORK COEFFICIENT, DEFLECTION COEFFICIENT, AND PRESSURE CO-EFFICIENT

Flow Coefficient: It is defined as the ratio of flow (or axial) velocity at inlet to the blade velocity. Mathematically, we can write

Flow coefficient,

Now $u = v_{f_1} (\tan \alpha_1 + \tan \beta_1)$

Also, $v_{f_1} = v_{f_2} = v_f$

Head or Work Coefficient: It is defined as the ratio of actual work done

to the kinetic energy corresponding to the mean peripheral velocity. One can write

Head or work coefficient,

Deflection Coefficient: ϕ_{def}: It is defined as,

Pressure coefficient: It is defined as the ratio of isentropic work done to kinetic energy corresponding to the peripheral velocity.

15.10 ❒ PRESSURE RISE IN A STAGE AND NUMBER OF STAGES

The static pressure ratio from Eq. (15.15) is,

Temperature rise in rotating blades,

Pressure rise in rotating blades,

Pressure rise in stationary blades is,

Pressure increase in stage is,

The stagnation pressure ratio is,

If the work done per stage is assumed to be the same, then number of stage z is given by,

If pressure ratio per stage is the same, then

The overall pressure ratio is,

Note that the pressure rise per stage in axial-flow compressor is less than that of a centrifugal compressor. This is because of no centrifugal action in axial flow compressor.

15.11 ❐ SURGING, CHOKING, AND STALLING

1. **Surging:** The delivery pressure *v*'*s* mass flow rate in an axial flow compressor is shown in Fig. 15.7. The mass flow is zero when the discharge valve is closed, but the air entrapped in the blade area is compressed and pressure of air increases. The condition is represented by point '*a*' in Fig. 15.7. As the valve is opened, the flow of air starts and the pressure rises upon point '*b*'.

   Any increase in air mass flow rate after point '*b*' is accompanied by a decrease in delivery pressure. This happens because the rate of increase in pressure loss due to friction is more than the rate of increase in pressure rise due to diffuser. Theoretically, the decrease in delivery pressure is continued up to point '*e*'. In practice, the maximum mass flow rate is limited by point '*c*' because beyond point '*c*', the mass flow exceeds design mass flow, the air angles are

widely different from vane angles, and choking takes place.

If the compressor is working between points ‘*a*’ and ‘*b*’, as the mass flow is decreased, the delivery pressure also decreases. However, the pressure in the downstream side of the compressor does not fall quickly, resulting in reversal of flow from downstream side towards the resulting pressure gradient. When this occurs, the pressure to the downstream side of the compressor falls and the compressor will start delivering air and the cycle would be repeated continuously. Thus, the flow would be pulsating between point ‘*a*’ and ‘*b*’. This pulsating air flow phenomena is known as ‘surging’. The surging causes overheating and stress reversal in the blades and damages the compressor. Within the region *bc*, the flow is stable. A fall in mass flow rate will result in rise in delivery pressure to restore to fall.

Figure 15.7 *Characteristic curve*

2. **Choking:** The maximum mass flow rate possible in a compressor is known as choking. The point ‘*e*’ in Fig. 15.7 represents the choking of the compressor.
3. **Stalling**: It is a phenomena which occurs on the suction side of the compressor in which the flow breaks away from the aerofoil blading. It may be due to lesser flow rate than the designed value or due to non-uniformity in the profile of the blades. Thus, stalling precedes surging. Stalling is a local phenomenon, whereas surging is a complete system phenomenon.

The relationship between pressure rise and mass flow rate at various speeds is shown in Fig. 15.8. At a certain speed, efficiency increases as the mass flow rate increases and reaches a maximum value after which it decreases, as shown in Fig. 15.9. The power consumed increases as mass flow increases. Figure 15.10 shows the performance and constant efficiency curves. The performance curves are plotted with dimensionless parameters, , and At constant value of there is a considerable narrower range of mass flow rate than in centrifugal compressor. The surge point is reached in axial compressors much before the maximum pressure ratio for given value of . Since the design usually

calls for the operating line to be near the maximum point on the curves, it follows that the operating line for axial flow compressors must be very near the surge line, thus narrowing the range of stable operation.

Figure 15.8 *Pressure ratio v's mass flow rate*

Figure 15.9 *Power and efficiency v's mass flow rate*

Figure 15.10 *Pressure rise v's volume flow*

15.13 ❐ COMPARISON OF AXIAL FLOW AND CENTRIFUGAL COMPRESSORS

The merits and demerits of axial flow compressors are given in Table 15.1.

Table 15.1 *Merits and demerits of axial flow compressor over centrifugal compressor*

15.14 ❐ APPLICATIONS OF AXIAL FLOW COMPRESSORS

Axial flow compressors have wide

applications in the following:

1. Jet engines
2. Gas turbines for power plants
3. Steel mills
4. Aircraft applications.

Note that the requirements of compressors for aircraft application:

1. High efficiency
2. Smaller frontal area
3. Suitability for multistaging
4. High starting torque
5. Lesser drag

15.15 ❐ LOSSES IN AXIAL FLOW COMPRESSORS

The following losses occur in axial flow compressors:

1. Profile losses on the surface of blades
2. Skin friction loss on the annulus walls
3. Secondary flow losses

Example 15.1

An axial compressor is fitted with half-reaction blading, the blade inlet and outlet angles being 50° and 15° when measured from the axial direction. The mean diameter of a certain blade pair is 85 cm and the rotor runs at 5500 rpm. Calculate the necessary isentropic efficiency of the stage if the pressure ratio of compression is to be 1.4 and the inlet air temperature is 25°C. Take c_p =1.005 kJ/kg. K, $\gamma = 1.4$.

[CSE, 1987]

Solution

$\beta_1 = 50°$, $\beta_2 = 15°$, $v_{f1} = v_{f2} = v_f$, $\alpha_1 = 15°$, $\alpha_2 = 50°$

Mean blade speed,

From the velocity triangles, (Fig. 15.11)

$$v_{w_1} = 0.1826\ u$$
$$v_{w_2} = 0.8164\ u$$
$$\Delta v_w = v_{w_2} - v_{w_1} = 0.6328\ u = 0.6328 \times 244.78 = 155 \text{ m/s}$$

Work done/kg, $w_a = u \times \Delta v_w = 244.78 \times 155 = 37940.9$ N.m/kg

Isentropic work, $w_{\text{isen}} = c_p\,(T_2 - T_1)$

Figure 15.11 *Velocity diagrams for axial flow compressor*

$$w_{\text{isen}} = 1.005\ (328 - 298) = 30.15 \text{ kJ/kg}$$

Isentropic efficiency,

Example 15.2

An axial flow compressor comprises a number of similar stages with equal work done per stage, and the velocity of flow is uniform throughout the compressor. The following data is given:

Overall stagnation pressure ratio = 4

Stagnation inlet temperature = 330 K

Relative air angle at rotor inlet = 130°

Relative air angle at rotor outlet = 100°

Blade velocity = 180 m/s

Degree of reaction = 0.5

Overall stagnation adiabatic efficiency = 0.86

Calculate (a) the stagnation outlet temperature, and (b) number of stages.

Solution

The velocity diagrams are shown in Fig. 15.12.

Stagnation adiabatic efficiency,

Figure 15.12 *Velocity diagrams for axial flow compressor*

$T_{02'} = 516.5$ K

For degree of reaction = 0.5, the velocity diagrams are similar.

$$\therefore u - v_{w2} = v_{w1}$$

$$v_{w1} = 0.1737\,u$$
$$v_{w2} = u - v_{w1} = 0.8263\,u$$
$$\Delta v_w = v_{w2} - v_{w1} = 0.6526\,u$$

Work done per stage = $u \cdot \Delta v_w$ = 180 × 0.6526 × 180 = 21144 N.m/kg

Total work = $c_p\,(T_{02} - T_{01})$ = 1.005(490.38 − 330) = 161.182 kJ/kg

Number of stages = = 7.62 = 8

Example 15.3

Air at a temperature of 300 K enters a 10-stage axial flow compressor at the rate of 3.5 m/s. The pressure ratio is 6.0 and the isentropic efficiency is 90%. The process is adiabatic and the compressor has symmetrical stages. The axial velocity is uniform across the stage and equals to 120 m/s and the mean blade speed of each stage is 200 m/s. Assume that the temperature change is same in each stage.

Determine the direction of the air at entry to and exit from the rotor and the stator blades. Also, find the power given to the air. Take c_p = 1.005 kJ/kg.K and $\gamma = 1.4$.

[IES, 2001]

Solution

The velocity diagrams are shown in Fig. 15.13.

$z = 10$, $T_1 = 300$ K, $\dot{m} = 3.5$ kg/s, $r_p = 6$, $\eta_i = 0.9$,
$v_f = 120$ m/s, $u = 200$ m/s

Figure 15.13 *Velocity diagrams for axial flow compressor*

Increase in temperature per stage,

$\tan \beta_1 = 1.2999$, $\beta_1 = 52.43°$, $\beta_2 = 20.14°$

Power supplied = $z\dot{m}c_p\,(\Delta T)_{\text{stage}}$ = $10 \times 3.5 \times 1.005 \times 22.284 = 783.84$ kW

Example 15.4

In an axial flow air compressor: u = 250 m/s, v_f = 200 m/s, $\alpha_1 = 50°$, $\alpha_2 = 15°$ and $\rho = 1$ kg/m^3. Determine (a)

the pressure rise, and (b) the work done per kg of air.

Solution

1. Pressure rise through a moving blade ring,
2. Work done per kg of air,

$$w = u\,(v_{w_1} - v_{w_2}) = u v_f\,(\tan \alpha_1 - \tan \alpha_2)$$
$$= 250 \times 200\,(\tan 50° - \tan 15°) \times 10^{-3} = 46.19 \text{ kW}$$

Example 15.5

An axial flow air compressor having eight stages and 50% degree of reaction compresses air in the pressure ratio of 4.5 : 1. Air enters the compressor at 27°C and flows through it with a constant speed of 100 m/s. The moving blades of compressor rotate with a mean speed of 200 m/s. The isentropic

efficiency of the compressor is 85%. Calculate (a) the work done on the compressor and (b) blade angles. Take $\gamma = 1.4$ and $c_p = 1.005$ kJ/kg.K for air.

Solution

$z = 8$, $R_d = 0.5$, $r_p = 4.5$, $T_1 = 273 + 27 = 300$ K, $v_f = 100$ m/s, $u = 200$ m/s, $\eta_{\text{isen}} = 0.85$

1. Work required per kg of air,

$$w = c_p\,(T_{2'} - T_1) = 1.005\,(489.4 - 300) = 190.36 \text{ kJ/kg}$$

2. Total work done $= zu\,(v_{w2} - v_{w1}) = zu\,v_f\,(\tan\alpha_2 - \tan\alpha_1)$

$$190.36 = 8 \times 200 \times 100\,(\tan\alpha_2 - \tan\alpha_1) \times 10^{-3}$$

$$\tan\alpha_2 - \tan\alpha_1 = 1.1897$$

For $R_d = 0.5$, $\alpha_1 = \beta_2$ and $\alpha_2 = \beta_1$

$\therefore\ 1.1897 = \tan\beta_1 - \tan\beta_2$

or $\tan\beta_1 - \tan\alpha_1 = 1.1897$

Now, $u = v_f(\tan \alpha_1 + \tan \beta_1)$

Adding the above two equations, we get

$$2 \tan \beta_1 = 3.1897$$

or $\tan \beta_1 = 1.5949$

or $\beta_1 = 57.9° = \alpha 2$

$$\tan \alpha_1 = \tan \beta_1 - 1.1897 = 0.4052$$

or $\alpha_1 = 22° = \beta_2$

Example 15.6

The overall isentropic efficiency of an axial flow air compressor is 83%. It draws air at 25°C and 1 bar and compresses to 4 bar. Assuming 50% degree of reaction, blade velocity 180 m/s, exit angle from stator (α_1) = 15°, inlet angle to rotor (β_1) = 45° and work input factor as 0.80,

calculate the flow velocity and the number of stages.

Solution

$\eta_{\text{isen}} = 0.83$, $T_1 = 273 + 25 = 298$ K, $p_1 = 1$ bar, $p_2 = 4$ bar, $R_d = 0.5$, $\alpha_1 = 15°$, $\beta_1 = 45°$, $u = 180$ m/s, $\phi_w = 0.80$

Work required per kg of air

$$= c_p (T_2' - T_1) = 1.005\,(472.5 - 298) = 175.37 \text{ kJ/kg}$$

From inlet velocity triangle (Fig.15.13), we have

For $R_d = 0.5$, $\alpha_2 = \beta_1 = 45°$ and $\alpha_1 =$

$$\beta_2 = 15°$$

$$v_{w2} = v_f \tan \alpha_2 = 141.96 \tan 45° = 141.96 \text{ m/s}$$
$$v_{w1} = v_f \tan \alpha_1 = 141.96 \tan 15° = 38.04 \text{ m/s}$$

Work done per kg per stage = $\phi_w u(v_{w2} - v_{w1})$

$$= 0.80 \times 180\ (141.96 - 38.04) \times 10^{-3} = 14.965 \text{ kJ/kg}$$

Number of stages, $z = = 11.72 = 12$

Example 15.7

In a 10-stage axial flow air compressor, the overall stagnation pressure ratio is 4 : 1 with an overall isentropic efficiency of 90%. The

inlet stagnation temperature and pressure are 293 K and 1 bar. The work is divided equally between the stages. The mean blade speed is 180 m/s and 50% reaction blading are used. The flow velocity throughout the compressor is 90 m/s. Calculate (a) the blade angles, and (b) the power required.

Solution

$z = 10$, $r_{op} = 4$, $\eta_{\text{isen}} = 0.9$, $T_{01} = 293$ K, $p_{01} = 1$ bar, $u = 180$ m/s, $R_d = 0.5$, $v_f = 90$ m/s

1. For $R_d = 0.5$, $\alpha_1 = \beta_2$ and $\alpha_2 = \beta_1$

   With isentropic compression, the temperature of air leaving the compressor is,

   Work required by the compressor = c_p ($T_{02'}$ −

T_{01})

$$= 1.005\ (451.2 - 293) = 159 \text{ kJ/kg}$$
$$= zu\ (v_{w2} - v_{w1})$$
$$= zuv_f\ (\tan \alpha_2 - \tan \alpha_1)$$

From inlet velocity triangle, we have,

Adding the above two equations, we get

$$2 \tan \beta_1 = 2.9815$$

or $\tan \beta_1 = 1.4908$

or $\beta_1 = 56.15° = \alpha_2$

$$\tan \alpha_1 = 2 - 1.4908 = 0.5092$$

or $\alpha_1 = 26.98° = \beta_2$

2. Power required = $\dot{m}c_p(T_{02'} - T_{01}) = 1 \times 1.005\ (451.2 - 293) =$ 159 kW

Example 15.8

The overall stagnation pressure ratio and isentropic efficiency for an axial flow air compressor are 5 and 82%,

respectively. The inlet stagnation pressure and temperature are 1 bar and 303 K. The mean blade speed is 200 m/s and degree of reaction is 0.5. The relative air angles are 12° and 30° at the rotor inlet and outlet, respectively. The work input factor is 0.9. Calculate (a) the stagnation polytropic efficiency, (b) the number of stages, (c) the inlet temperature and pressure, and (d) the blade height in the first stage if the hub-tip ratio is 0.42 and mass flow rate is 20 kg/s.

Solution

$r_{op} = 5$, $\eta_{\text{isen}} = 0.82$, $p_{01} = 1$ bar, $T_{01} = 303$ K, $u = 200$ m/s, $R_d = 0.5$, $\beta_1 = 30° = \alpha_2$, $\beta_2 = 12° = \alpha_1$, $\phi_w = 0.9$, $\dot{m}$

= 20 kg/s, hub-tip ratio = 0.42.

1. Polytropic efficiency,

2. From the inlet velocity triangle (Fig. 15.13), we have

$$v_{w2} = v_{f2} \tan \alpha_2 = 253.2 \tan 30° = 146.2 \text{ m/s}$$

$$v_{w1} = v_f \tan \alpha_1 = 253.2 \tan 12° = 53.8 \text{ m/s}$$

Work consumed per stage = $\phi_w\, u(v_{w2} - v_{w1})$

$$= 0.9 \times 200\ (146.2 - 53.8) \times 10^{-3} = 16.632 \text{ kJ/kg}$$

Total work consumed by the compressor = c_p $(T_{0z'} - T_{01})$

$$= 1.005\ (518.7 - 303) = 216.78 \text{ kJ/kg}$$

Number of stages,

3. = 258.86 m/s

Static temperature,

4. = 0.859 kg/m^3

$$m = \rho_1 A_1 v_f$$

or $20 = 0.859 \times \pi\, [1 - (0.42)^2] \times 253.2$

or $r_1 = 0.1885$ m or 188.5 mm

Example 15.9

A multistage axial flow air compressor delivers 25 kg/s of air. The inlet stagnation condition is 1 bar and 20°C. The power consumed by the compressor is 4500 kW. Calculate (a) the delivery pressure, (b) the number of stages, and (c) the overall isentropic efficiency of the compressor.

Assume that temperature rise in the first stage is 15° C, the polytropic efficiency of compression is 0.90 and the stage stagnation pressure ratio is constant.

Solution

$\dot{m} = 25$ kg/s, $p_{01} = 1$ bar, $T_{01} = 273 + 20 = 293$ K, $P = 4500$ kW, $T_{02} = 293 + 15 = 308$ K, $n_p = 0.90$

1. Polytropic efficiency

   Power $P = \dot{m}c_p\,(T_{0z'} - T_{01})$

   or $4500 = 25 \times 1.005\,(T_{0z'} - 293)$

   or $T_{0z'} = 472.1$ K

   Again, polytropic efficiency can be expressed as

2.
3.

Example 15.10

An axial compressor receives 1000 m^3/min of free air at 15°C and 0.9 bar. The blades are of aerofoil type having projected area and blade

length as 19.25 cm^2 and 6.75 cm, respectively. The blade ring mean diameter is 60 cm and the speed is 6000 rpm. On each blade ring, there are 50 blades and the blades occupy 10% of the axial area of flow. Value of C_L and C_D are 0.6 and 0.05 respectively at zero angle of incidence. Assuming isentropic compression, calculate the pressure rise per blade ring and the power input per stage. Assume axial inlet.

Solution

$\dot{V} = 1000\ m^3/min$, $T_1 = 273 + 15 = 288$ K, $p_1 = 0.9$ bar, $A = 19.25\ cm^2$, $h = 6.75$ cm, $k_b = 0.9$, $C_L = 0.6$, $C_D = 0.05$, $D_m = 60$ cm, $i = 50$

Density of air,

Area across flow, $A_f = k_b \pi D_m h$

$$= 0.9 \times \pi \times 0.6 \times 6.75 \times 10^{-2}$$
$$= 0.1145 \text{ m}^2$$

Blade velocity,

For axial inlet, $v_{a1} = v_f = 145.55$ m/s

The velocity triangle at inlet is shown in Fig. 15.14.

Figure 15.14 *Velocity diagram for axial flow compressor*

Power input per stage,

$$p = (F_L \cos \beta_1 + F_D \sin \beta_1)\, ui$$
$$= (35.7 \cos 52.325° + 2.975 \sin 52.325°) \times 188.5 \times 50 \times 10^{-3}$$

$= 227.839$ kW

Mass flow rate,

$p_2 = 0.9 \times 1.16 = 1.044$ bar

Pressure rise $= p_2 - p_1 = 1.044 - 0.9$
$= 0.144$ bar

Example 15.11

An axial flow compressor gives a pressure rise of 4 : 1 and the total head isentropic efficiency is 86%. Stagnation inlet temperature is 17°C. The inlet and outlet air angles from the rotor blades are 45° and 10°, respectively. The rotor and stator blades are symmetrical. The mean blade and axial velocity remain constant. Assuming blade

speed of 220 m/s and work input factor 0.86, find the polytropic efficiency, number of stages required and Mach number. Take R = 287 J/kg.K.

Solution

$r_p = 4$, $\eta_{o\ \mathrm{isen}} = 0.86$, $T_{01} = 273 + 17 = 290$ K, $\alpha_1 = 45°$, $\alpha_2 = 10°$, $u = 220$ m/s, $\phi_w = 0.86$

Polytropic efficiency,

Head isentropic efficiency,

The velocity diagrams are shown in

Fig. 15.15.

$$\alpha_1 = \beta_2 = 10°, \alpha_2 = \beta_1 = 45°$$
$$u = v_{f1} (\tan \alpha_1 + \tan \beta_1)$$

Power required per kg of air = $c_p (T_2 - T_1) = \phi_w \, u v_f (\tan \alpha_2 - \tan \alpha_1)$

Figure 15.15 *Velocity diagrams at inlet and outlet of vane*

Also,

$$T_{02} = 290 \times 1.565 = 453.86 \text{ K}$$

Total temperature rise = $T_{02} - T_{01}$

$$= 453.86 - 290 = 163.86°\text{C}$$

Number of stages,

From the inlet velocity diagram,

Example 15.12

Air at 1.01325 bar and 288 K enters an axial flow compressor stage with an axial velocity of 150 m/s. There are no inlet guide vanes. The rotor stage has a tip diameter of 60 cm and a hub diameter of 50 cm and rotates at 100 rps. The air enters the rotor and leaves the stator in the axial direction with no change in velocity or radius. The air is turned through 30° as it passes through rotor.

Assuming constant specific heat and that air enters and leaves the blade at the blade angles, construct the

velocity diagrams at mean diameter for this stage, and calculate (a) the mass flow rate, (b) the power required, and (c) the degree of reaction.

Solution

$p_1 = 1.01325$ bar, $T_1 = 288$ K, $v_f = 150$ m/s, $D_2 = 60$ cm, $D_1 = 50$ cm, $N = 100$ rps, $\beta_2 - \beta_1 = 30°$

The velocity triangles are shown in Fig. 15.16.

Figure 15.16 *Velocity diagrams at inlet and outlet of vane*

or $\beta_1 = 49.04°$

$$\beta_2 - \beta_1 = 30° \text{ or } \beta_1 - \beta_2 = 30°$$

$$\text{or } \beta_2 = 19.04° \text{ or } 79.04°$$

$$v_{f2} \tan \beta_2 = u + v_{w2}$$
$$150 \tan 19.04° = 51.77 < 172.8 \text{ m/s}$$

$$\therefore v_{w2} \text{ is } -\text{ve}$$

Volume flow rate,

1. Mass flow rate,

$$\dot{m} = \dot{V}\rho = 12.96 \times 1.11 = 14.384 \text{ kg/s}$$

2. Power required,

$$P = \dot{m}uv_f = 14.384 \times 172.8 \times 150 \times 10^{-3} = 372.85 \text{ kW}$$

3. Degree of reaction,

Example 15.13

An axial flow compressor with pressure compression ratio 4 draws air at 20°C and delivers at 197°C. The mean blade speed and flow velocity are constant throughout the

compressor. Assuming 50% reaction and blading velocity as 180 m/s, find the flow velocity and number of stages. Take work input factor = 0.82, $\alpha_1 = 20°$, $\beta_1 = 42°$, $c_p = 1.005$ kJ/kg K.

Solution

Given: $r_p = 4$, $T_1 = 273 + 20 = 293$ K, $T_2 = 273 + 197 = 470$ K, $u = u_1 = u_2$ and $v_{f1} = v_{f1} = v_f$, $R_d = 0.5$, $u = 180$ m/s, $\phi_w = 0.82$, $\alpha_1 = 20°$, $\beta_1 = 42°$, $c_p = 1.005$ kJ/kg K

For 50% degree of reaction,

$$\alpha_2 = \beta_1 = 42° \text{ and } \beta_2 = \alpha_1 = 20°$$

Work input factor,

Refer to Fig. 15.17

$$\Delta v_w = v_{w2} - v_{w1} = v_f (\tan \alpha_2 - \tan \alpha_1)$$
$$= v_f (\tan 42° - \tan 20°) = 0.5364\ v_f$$

$$\therefore 0.5364\ v_f = 73.8$$

Flow velocity, $v_f = 137.6$ m/s

Work done per stage $= u \times \Delta v_w$

$= 180 \times 73.8 = 13284$ Nm/kg

Total work $= c_p (T_2 - T_1)$

$= 1.005 \times 10^3 (470 - 293) = 177885$ Nm/kg

Number of stages,

$= 13.39 \simeq 14$

Figure 15.17 *Velocity diagrams for axial flow compressor*

Example 15.14

An eight-stage axial flow compressor provides an overall pressure ratio of 6: 1 with an overall isentropic efficiency 90% when the temperature of air at inlet is 20°C. The work divided equally between the stages. A 50% reaction design is used with a mean blade speed 188 m/s and a constant axial velocity 100 m/s through the compressor. Estimate the power required and the blade angles. Take $c_p = 1.005$ kJ/kg.K and $\gamma = 1.4$.

Solution

Given: $z = 8$, $p_z/p_1 = 6$, $(\eta_{\text{isen}})_0 = 0.9$, $T_1 = 273 + 20 = 293$ K, $\mathrm{R}_d = 0.5$, $u = 188$ m/s, $v_f = 100$ m/s

Refer to Fig. 15.18.

For $R_d = 0.5$, $\alpha_1 = \beta_2$, $\alpha_2 = \beta_1$

From Fig. 15.19, we have $v_{w1} = v_{f1} \tan \alpha_1$ and $v_{w2} = v_{f2} \tan \alpha_2$

Work done per kg of air, $w = zu\,(v_{w2} - v_{w1})$

$$= 150.4\,(\tan \alpha_2 - \tan \alpha_1)$$
$$= 150.4\,(\tan \beta_1 - \tan \alpha_1)$$
$$= c_p\,(T_{z'} - T_1) = 0.24 \times 4.187\,(510.8 - 293) = 218.86$$

Figure 15.18 *T-s diagram*

Figure 15.19 *Velocity triangles*

$$\therefore \tan \beta_1 - \tan \alpha_1 = 1.455$$

After solving, we get

$$2 \tan \beta_1 = 3.335$$

$$\text{or } \beta_1 = 59°$$

$$\tan \alpha_1 = 1.88 - 1.6676 = 0.2124$$

$$\text{or } \alpha_1 = 12°$$

$$\text{Power required} = c_p\,(T_z' - T_1) = 0.24 \times 4.187\,(510.8 - 293)$$

$$= 218.86 \text{ kW}$$

Example 15.15

The first stage of an axial compressor is designed on free vortex principles with no inlet guide vanes. The rotational speed is 6000 rpm and the stagnation temperature rise is 20 K. The hub-tip ratio is

0.60, work done factor is 0.93 and isentropic efficiency of the stage is 0.89. Assuming an inlet velocity of 140 m/s and ambient conditions of 1.01 bar and 288 K, calculate (a) the tip radius and corresponding rotor air angles, if the Mach number relative to the tip is limited to 0.95, (b) the mass flow entering the stage, (c) the stage stagnation pressure ratio and power input, and (d) rotor air angles at the root section. Assume c_p = 1.005 kJ/kgK and γ = 1.4.

Solution

Given: N = 6000 rpm, = 20 K, = 0.6, ϕ_w = 0.93, $(\eta_{isen})_{stage}$ = 0.89, v_{a1} = 140 m/s, p_1 = 1.01 bar, T_1 = 288

K, M = 0.95, c_p = 1.005 kJ/kg.K, γ = 1.4

With no inlet guide vanes, $\alpha_1 = 0°$ and $v_{f1} = v_{a1} = 140$ m/s, $v_{w1} = 0$

For free vortex flow = $v_{w1} \times r_h = v_{w2} \times r_t$

The velocity diagrams are as shown in the Fig. 15.20(a).

Also, $v_{f1} = v_{f2} = v_f$

$\therefore v_{a2} = v_{a1}$ and $\beta_1 = \beta_2$

Figure 15.20 *(a) Velocity diagrams, (b) T-s diagram for axial flow compressor*

$$T_{02} = T_{01} \,(T_{02'} - T_{01}) \times (\eta_{\text{isen}})_{\text{stage}}$$

$= 297.75 + 20 \times 0.89 = 315.55$ K

1. Mach number relative to tip, M = 0.95

   or $v_{r2} = 333.1$ m/s

   Tip diameter, $D_t = 962$ mm

   Hub diameter, $D_h = 0.6\ D_t = 0.6 \times 962 = 577.2$ mm

2. Density of air at inlet,

   Mass flow rate of air entering the stage,

3.
4. $\alpha_1 = \alpha_2 = 0°$, $\beta_1 = \beta_2 = 65.14°$

Example 15.16

An axial flow compressor has an overall pressure ratio of 4 and mass flow rate of 3 kg/s. If the polytropic efficiency is 88% and the stagnation temperature rise per stage must not

exceed 25 K, calculate the number of stages required and pressure ratio of first and last stages. Assume equal temperature rise in all stages. If the absolute velocity approaching the last rotor is 165 m/s at an angle of 20° from the axial direction, the work done factor is 0.83, the velocity diagram is symmetrical and the mean diameter of last stage rotor is 18 cm, calculate the rotational speed and length of the last stage rotor blade inlet to the stage. Ambient conditions are 1.01 bar and 288 K.

Solution

Given: = 4, $\dot{m}_a$ = 3 m/s, $\eta_p = 0.88$, $(dT_0\eta)_{stage} \leq 25K$, $(v_{a1})_z = 165$ m/s,

$$\alpha_1 = 20°,\ \phi_w = 0.83,\ (d_m)_z = 0.18 \text{ m},$$
$$p_1 = 1.01 \text{ bar},\ T_1 = 288 \text{ K}$$

Figure 15.21 *T – s diagram*

Refer to Fig. 15.21.

$$\text{Now, } (v_{a1})_1 = (v_{a1})_z = 165 \text{ m/s}$$

Pressure ratio of first and last stage,

$$\text{or } n = 1.48$$

$$\therefore T_{0z'} = 301.5(3.8)^{0.48/1.48} = 464.9 \text{ K}$$

Number of stages,

Actual stagnation temperature rise per stage

Hence O.K.

For symmetrical velocity diagram, $\alpha_1 = \beta_2$ (see Fig. 15.22)

Work done per stage, $w = c_p \times (dT'_0)_{stage} \times \phi_w$

$$= 1.005 \times 23.4 \times 0.83 = 19.519 \text{ kJ/kg}$$

Also, $w = (v_{w_2} - v_{w_1})u$

$$= (v - 2v_{w_1})u = (u - 2 \times v_{a_1} \sin \alpha_1)u$$
$$= (u - 2 \times 165 \sin 200)u = (u - 112.87)u$$

or $u^2 - 112.87\ u = 19519$

or $u^2 - 112.87\ u - 19519 = 0$

Figure 15.22 *Symmetrical velocity diagrams*

Area across flow, $A_f = \pi d_m h$

Density of air, $\rho_1 = = 1.222$ kg/m^3

Velocity of flow, $v_{f1} = v_{a1} \cos \alpha_1 = 165 \cos 20° = 155$ m/s

$$\dot{m}_a = \rho_1 A_f v_{f1}$$
$$3 = 1.222 \times \pi \times 0.18 \times h \times 155$$
$$h = 0.028 \text{ m or } 2.8 \text{ cm}$$

Length of last stage rotor blade, $h = 2.8$ cm.

Example 15.17

An axial-flow compressor employed in gas turbine plant delivers air at the rate of 300 kg/s and develops a

total pressure ratio of 20. The inlet stagnation conditions are 1 bar and 300 K and the blade speed is kept at 200 m/s to minimise noise generation. The stage degree of reaction at the mean blade height is 50%. The axial velocity of flow is 160 m/s. The work done factor is 0.88. The hub-tip diameter ratio is 0.8. Assume actual temperature rise in each stage. Show the process on *T-s* diagram and draw velocity diagram. Find (a) all the fluid angles of the first stage and (b) The hub and tip diameter including blade height. Take R = 0.287 kJ/kg-K and c_p = 1.005 kJ/kg-K.

[IES, 2010]

Solution

Given: $\dot{m}_a = 300$ kg/s, $r_p = 20$, $T_{01} = 300$ K, $p_{01} = 1$ bar, $\eta_{isen} = 0.87$, $z = 18$, $u = 200$ m/s, $R_d = 0.5$, $\phi_w = 0.88$, hub-tip ratio = 0.8, $R = 0.287$ kJ/kg-K, $c_p = 1.005$ kJ/kg-K, $v_f = 160$ m/s

The $T - s$ and velocity diagrams are shown in Fig. 15.23.

1. **Figure 15.23** *(a) T-s diagram, (b) Velocity diagrams*

$$T_{oz'} = 766.7 \text{ K}$$
$$v_{w2} = v_f \tan \alpha_2,\ v_{w1} = v_f \tan \alpha_1$$

For $R_d = 0.5$, $\alpha_2 = \beta_1$, $\alpha_1 = \beta_2$

Work consumed by compressor per kg of air

$$= u\,(v_{w2} - v_{w1})\,\phi_w \times z$$
$$= c_p\,(T_{oz'} - T_{01})$$

200 (tan α_2 – tan α_1) 160 × 0.88 × 18 = 1.005 (766.7 – 300)

$$\tan \alpha_2 - \tan \alpha_1 = 9.25 \times 10^{-4}$$

or $\tan \beta_1 - \tan \alpha_1 = 9.25 \times 10^{-4}$

Adding the above two equations, we get

$$2 \tan \beta_1 = 1.250925$$

or $\beta_1 = 32.02° = \alpha_2$

or $\tan \alpha_1 = 1.25 - 0.62546 = 0.62453$

$$\alpha_1 = 31.98° = \beta_2$$

2. Density of approaching air to first stage.

Applying continuity equation,

$$\rho_1 A_f v_f = \dot{m}_a$$

$1.1614 \times \pi [1 - (0.8)^2] \times 160 = 300$

Tip radius, $r_1 = 1.194$ m

Hub radius, $r_h = 0.8\ r_1 = 0.8 \times 1.194 = 0.9558$ m

Blade height of first stage,

$h = r_1 - r_h = 1.194 - 0.9558 = 0.238$ m.

Example 15.18

Air enters an axial flow compressor at 25°C and undergoes a pressure increase of six times that at inlet. The mean velocity of rotor blades is 200 m/s. The inlet and exit angles of both the moving and fixed blades are 45° and 20° respectively. The degree of reaction at the mean diameter is 50 percent and there are 10 stages in the compressor. The isentropic efficiency of compressor is 85% and axial velocity is constant throughout. Calculate the work done factor of the compressor.

Solution

Given: $T_1 = 273 + 25 = 298$ K, $r_p = 6$, $u_m = 200$ m/s, $\beta_1 = 45°$, $\beta_2 = 20°$, $R_d = 50\%$, $z = 10$, $(\eta_c)_{\text{isen}} = 85\%$, v_{f1}

$$= v_{f2} = v_f$$

For $R_d = 50\%$, $\alpha_2 = \beta_1$ and $\alpha_1 = \beta_2$

Also $v_{r1} = v_{a2}$ and $v_{r2} = v_{a1}$

Figure 15.24 *Velocity triangles*

From Fig 15.24, we have

$$\begin{aligned} u_m &= v_{r2} \sin \beta_2 + v_{a2} \sin \alpha_2 \\ &= v_{a1} \sin \beta_2 + \text{va1} \cos \alpha_1 \tan \alpha_2 \\ 200 &= v_{a1} [\sin 20° + \cos 20° \tan 45°] \\ &= 1.2817\ v_{a1} \end{aligned}$$

or $v_{a1} = 156$ m/s

$$\begin{aligned} v_{w1} &= v_{a1} \sin \alpha_1 = 156 \sin 20° = 53.36 \text{ m/s} \\ v_{w2} &= v_{a2} \sin \alpha_2 = 207.3 \sin 45° = 146.58 \text{ m/s} \\ \Delta v_w &= v_{w2} - v_{w1} = 146.58 - 53.36 = 93.22 \text{ m/s} \end{aligned}$$

Work done on compressor, $w_c = c_p \times \Delta T = 1.005 \times 234.35 = 235.5$ kJ/kg

Power input to compressor = $\Delta v_w \times$

$$u_m \times \phi_w \times z$$

$$= 93.22 \times 200 \times \phi_w \times 10 = 186,440 \times \phi_w$$

Work done factor,

Example 15.19

An axial flow compressor compresses the air up to overall stagnation pressure 10 bar with overall stagnation isentropic efficiency of 88%. The inlet stagnation pressure and temperature are 1 bar and 300 K. The mean blade speed is 200 m/s. The degree of reaction is 0.5 at the mean radius with air angles of 30° and 10° at rotor inlet and outlet with axial direction respectively. The work

done factor is 0.88. The hub-tip ratio is 0.4. The mass flow rate is 50 kg/s. Show the compression process on *T*-*s* diagram and draw the inlet and outlet velocity triangles. Find (a) the stagnation polytropic efficiency, (b) the number of stages and (c) the blade height in first stage of the compressor.

[IAS, 2006]

Solution

Given: $p_1 = 1$ bar, $T_{01} = 300$ K, $p_{oz} =$ 10 bar, $\eta_{\text{isen}} = 88\%$, $u_m = 200$ m/s, $R_d = 0.5$, $\beta_1 = 30°$, $\beta_2 = 10°$, $\phi_w =$ 0.88, , $\dot{m} = 50$ kg/s,

The *T*-*s* diagram is shown in Fig. 15.25(a)

The velocity triangles are shown in Fig.15.25(b).

1. Stagnation polytropic efficiency,

$$= 0.9117 \text{ or } 91.17\%$$

2. For $R_d = 0.5$

Figure 15.25 *(a) T-s diagram, (b) Velocity triangles*

$$v_{w2} = v_f \tan \alpha_2 = 265.4 \tan 30° = 153.2 \text{ m/s}$$
$$v_{w1} = v_f \tan \alpha_1 = 265.4 \tan 10° = 46.8 \text{ m/s}$$

Work consumed per stage = $\phi_w\, u\, (v_{w2} - v_{w1})$

$$= 0.88 \times 200\,(153.2 - 46.8) \times 10^{-3} = 18.726 \text{ kJ/kg}$$

Total work consumed by compressor = $c_p\,(T_{oz'} - T_{01})$

$$= 1.005\,(617.3 - 300) = 318.89 \text{ kJ/kg.}$$

Number of stages,

3. Static temperature,

$$\dot{m}_a = \rho_1 A_1 v_f$$
$$50 = 0.843 \times \pi\,[1 - (0.4)^2] \times 265.4$$
$$r_1 = 0.291 \text{ m or } 291 \text{ mm}$$
$$r_h = 0.4\, r_1 = 0.4 \times 291 = 116.4 \text{ mm}$$

Example 15.20

An air compressor has eight stages of equal pressure ratio 1.35. The flow rate through the compressor and its overall efficiency are 50 kg/s and 82 per cent respectively. If the condition of air at entry are 1.0 bar and 40°C, determine:

1. the state of air at the compressor exit,
2. polytropic or small stage efficiency,
3. efficiency of each stage,
4. power required to drive the compressor assuming overall efficiency of the drive as 90%

Take $c_p = 1.005$ kJ/kg K and $\gamma = 1.4$.

[IAS, 2005]

Solution

Given: $z = 8$, $(r_p)_{st} = 1.35$, $\dot{m} = 50$ kg/s, $\eta_0 = 82\%$, $p_1 = 1.0$ bar, $T_1 = 313$ K, $\eta_{mech} = 90\%$, $c_p = 1.005$ kJ/kg.K, $\gamma = 1.4$

1.

Exit pressure, $p_a = 1.0 \times 11.03 = 11.03$ bar

2. Polytropic efficiency,
3.

$3.0415\, n - 3.0415 = n$

or $n = 1.49$

Isentropic efficiency of each stage

4. Power required to drive the compressor,

Example 15.21

A small compressor has the following data:

Air flow rate = 1.5778 kg/s

Pressure ratio = 1.6

Rotational speed = 54,000 rpm

Efficiency = 85%

State of air at entry, p_{0p} = 1.008 bar, T_{0p} = 300 K

c_p for air = 1.009 kJ/kg K.

1. Calculate the power required to drive this compressor.
2. A geometrically similar compressor of three times the size is constructed. Determine, for this compressor (i) mass flow rate, (ii) pressure ratio, (iii) speed, and (iv) the power required.

Assume same entry conditions and efficiency for the two compressors and also assume kinematic and dynamic similarities between the two machines.

[IAS, 2005]

Solution

Given: $\dot{m}_a$ = 1.5778 kg/s, r_p = 1.6, N = 54,000 rpm, η = 85%, p_{01} = 1.008 bar, T_{01} = 300 K, c_p = 1.009 kJ/

kg.K, $D_2 = 3D_1$

1. Power required = $\dot{m}_a c_p (T_{02'} - T_{01})$ = 1.5778 × 1.009 (350.725 − 300) = 80.75 kW
2. = const., G = mass flow rate.
    1.
    2.
    3.
    4. p_2 = 21.24 × 1.009 (350.725 − 300) = 1087.6 kW

Example 15.22

Air at a temperature of 300 K enters a 10-stage axial flow compressor at the rate of 3.5 kg/s. The pressure ratio is 6.0 and the isentropic efficiency is 90%. The process is adiabatic and the compressor has symmetrical stages. The axial velocity of 120 m/s is uniform across the stages and the mean blade speed is 200 m/s. Assume that the temperature change is same in each

stage.

Determine the direction of the air at entry to and exit from the rotor and stator blades. Also find the power given to the air.

For air, take $c_p = 1.005$ kJ/kg.K and $\gamma = 1.4$.

[IAS, 2004]

Solution

Given: $T_1 = 300$ K, $z = 10$, $\dot{m}_a = 3.5$ kg/s, $r_p = 6$, $\eta_{\text{isen}} = 90\%$, $v_f = 120$ m/s, $u = 200$ m/s, $c_p = 1.005$ kJ/kg.K, $\gamma = 1.4$

Symmetrical stages, same temperature change in each stage

The velocity diagrams are shown in Fig. 15.26

$$(\Delta T)_{\text{stage}} = (\Delta T)_{\text{overall}}/z$$

Increase in temperature per stage,

Figure 15.26 *Velocity triangles*

$\tan \beta_1 = 1.2999$, $\beta_1 = 52.43°$, $\beta_2 = 20.14°$

$$\text{Power supplied} = z\dot{m}_a c_p (\Delta T)_{\text{stage}}$$

$$= 10 \times 3.5 \times 1.005 \times 22.284$$

$$= 783.84 \text{ kW}$$

Example 15.23

An axial flow compressor of 50% reaction blading has isentropic efficiency of 82%. It draws air at 17°C and compresses in the pressure ratio of 4 : 1. The mean blade speed and flow velocity are constant throughout the compressor. The inlet and outlet angles of blades are 15° and 45° respectively (angles measured from axial direction). Blade speed = 180 m/s and work input factor = 0.84. Calculate (a) the flow velocity and (b) the number of stages. **[IAS, 2002]**

Solution

Given: $R_d = 50\%$, $\eta_{\text{isen}} = 82\%$, $T_1 = 17 + 273 = 290$ K, $r_p = 4{:}1$, $u_m =$ const, $\mathrm{v}_f =$ const., $\beta_1 = 15°$, $\beta_2 = 45°$,

$$u = 180 \text{ m/s}, \phi_h = 0.84$$

$$\text{For } R_d = 50\%, \alpha_1 = \beta_2 = 45°, \alpha_2 = \beta_1 = 15°$$

1. From Fig. 15.27, we get

$\therefore$ Flow velocity, $v_f = 142$ m/s

$w_{\text{isen}} = c_p (T_2 - T_1) = 1.005(430.9 - 290) = 141.6$ kJ/kg

Figure 15.27 *Velocity triangles*

2. Work done per stage = $u \times \Delta v_w = 180 \times 75.6 = 13.608$ kJ/kg

Number of stages

Summary for Quick Revision

1. In axial flow compressors, the air flows parallel to its axis throughout.
2. Axial flow compressors are particularly suitable for supercharging of internal combustion engines, gas turbines, and turbojets.
3. Aerofoil theory:
    1. Lift force,
    2. Drag force,
4. Power input per stage,

$V' =$ FAD in m^3/min.

5. Velocity relations:
    1.
6.
    1.
7. where α_1 = exit angle from stator, β_1 = inlet angle to rotor, α_2 = inlet angle to stator and β_2 = outlet angle from rotor.
8. Work required per stage per kg of air, $w = (v_{w2} - v_{w1})u = c_p (T_{02'} - T_{01})$

   $= v_f (\tan \alpha_2 - \tan \alpha_1)u$

   $= [u - v_f (\tan \alpha_1 + \tan \beta_2]u$
9. Stage isentropic efficiency,
10. Stagnation pressure ratio,
11.
12. For $R_d = 50\%$, the blades are symmetrical, i.e.,

    $\alpha_1 = \beta_2$ and $\alpha_2 = \beta_1$
13. Isentropic pressure rise through a cascade,
    1.
14. Polytropic efficiency: It is the small stage efficiency $(\eta_{\text{isen}})_{\text{stage}}$, which is constant for all stages, and is denoted by η_p.
    1.
    2.
15. Flow coefficient,
    1.
16. Head or work coefficient,
    1.
17. Pressure coefficient,
    1.
18. Number of stages,
    1. For same pressure ratio per stage,
19.
20. The pulsating air flow phenomenon is known as *surging*. It

causes overheating and stress reversal in the blades which damages the blades.

21. The maximum mass flow rate possible in a compressor is known as *choking*.
22. *Stalling* is a phenomenon in which the flow breaks away from the aerofoil blading. It occurs due to lesser flow rate than the designed value or due to non-uniformity of blade profile.

Multiple-choice Questions

1. A multistage compressor is to be designed for a given flow rate pressure ratio. If the compressor consists of axial flow stages followed by centrifugal instead of only axial flow stages, then the
    1. overall diameter would be decreased
    2. overall diameter would be increased
    3. axial length of the compressor would be increased
    4. axial length of the compressor would be decreased
2. What is the ratio of the isentropic work to Euler's work known as?
    1. Pressure coefficient
    2. Slip factor
    3. Work factor
    4. Degree of reaction
3. In air-craft gas turbines, the axial flow compressor is preferred because
    1. of high pressure rise
    2. it is stall-free
    3. of low frontal area
    4. of higher thrust
4. In axial flow compressor, exit flow angle deviation from the blade angle is a function of
    1. blade camber
    2. space-chord ratio
    3. both blade camber and space-chord ratio
    4. blade camber and incidence angle
5. High positive incidence in an axial compressor blade row leads to
    1. suppression of separation of flow on the blade
    2. choking of the flow
    3. separation of flow on the pressure side of the blade
    4. separation of flow on the suction side of the blade

6. Which one of the following is the correct expression for the degree of reaction for an axial-flow air compressor?
    1.
    2.
    3.
    4.
7. Which one of the following types of compressors is mostly used for supercharging of IC engines?
    1. Radial flow compressor
    2. Axial flow compressor
    3. Roots blower
    4. Reciprocating compressor
8. Phenomenon of choking in compressor means
    1. no flow of air
    2. fixed mass flow rate regardless of pressure ratio
    3. reducing mass flow rate with increase in pressure ratio
    4. increased inclination of the chord with air steam
9. Degree of reaction in an axial compressor is defined as the ratio of static enthalpy rise in the
    1. rotor to static enthalpy rise in the stator
    2. stator to static enthalpy rise in the rotor
    3. rotor to static enthalpy rise in the stage
    4. stator to static enthalpy rise in the stage
10. The usual assumption in elementary compressor cascade theory is that
    1. axial velocity through the cascade changes
    2. for elementary compressor cascade theory, the pressure rise across the cascade is given by equation of state
    3. axial velocity through the cascade does not change
    4. with no change in axial velocity between inlet and outlet, the velocity diagram is formed
11. Match List 1 with List II (pertaining to blower performance) and select the correct answer using the codes given below the Lists:

Codes:

A B C

1. 4 3 2
2. 1 3 2
3. 4 1 3
4. 2 3 4

12. The critical value of Mach number for a subsonic airfoil is associated with a sharp increase in drag due to local shock formation and its interaction with the boundary layer. A typical value of this critical Mach number is of the order of
    1. 0.4 to 0.5
    2. 0.75 to 0.85
    3. 1.1 to 1.3
    4. 1.5 to 2.0
13. The inlet and exit velocity diagrams of a turbomachine rotor are shown in Fig. 15.28.

Figure 15.28

The turbomachine is

1. an axial compressor with radial blades
2. a radial compressor with radial blades
3. a radial compressor with curved blades
4. an axial compressor with forward curved blades

14. The turbomachine used to circulate refrigerant in large refrigeration plant is
    1. A centrifugal compressor
    2. A radial turbine
    3. An axial compressor
    4. An axial turbine
15. The energy transfer process is
    1. continuous in a reciprocating compressor and intermittent in an axial compressor
    2. continuous in an axial compressor and intermittent in a reciprocating compressor
    3. continuous in both reciprocating and axial compressors.
    4. intermittent in both reciprocating and axial compressors
16. In an axial flow compressor stage, air enters and leaves the stage axially. If the whirl component of the air leaving the rotor is half the mean peripheral velocity of the rotor blades,

then the degree of reaction will be

1. 1.00
2. 0.75
3. 0.50
4. 0.25

17. If an axial flow compressor is designed for a constant velocity through all stages, the area of the annulus of the succeeding stages will
    1. remain the same
    2. progressively decrease
    3. progressively increase
    4. depend upon the number of stages
18. If the static temperature rise in the rotor and stator, respectively, are ΔT_A and $\Delta T_{B,}$ the degree of reaction in an axial flow compressor is given by
    1.
    2.
    3.
    4.
19. Which one of the following pairs of features and compressors type is NOT correctly matched?
    1. Intake and delivery ports compression is attained by back flow and internal compression cylindrical rotor set to eccentric casing: Vane compressor
    2. Intermittent discharge requires receiver, produces high pressure, slow speed and lubrication problems : Reciprocating compressor
    3. Continuous flow, radial flow, handles large volume, much higher speed and fitted into design of aero-engines : Centrifugal compressor
    4. Successive pressure drops through contracting passages, blades are formed from a number of circular arcs, axial flow : Axial flow compressor
20. In an axial flow compressor design, velocity diagrams are constructed from the experimental data of aerofoil cascades. Which one of the following statements in this regard is correct?
    1. Incidence angle of the approaching air is measured from the trailing edge of the blade
    2. δ is the deviation angle between the angle of incidence and tangent to the camber line
    3. The deflection ε of the gas stream while passing through the cascade is given by $\varepsilon = \alpha_1 - \alpha_2$

4. ε is the sum of the angle of incidence and camber less any deviation angle, i.e., $\varepsilon = i + \theta - \delta$

21. Consider the following statements regarding the axial flow in an air compressor:
    1. Surging is a local phenomenon while stalling affects the entire compressor.
    2. Stalling is a local phenomenon while surging affects the entire compressor.
    3. The pressure ratio of an axial compressor stage is smaller than that of a centrifugal compressor stage

    Of these statements

    1. (i), (ii) and (iii) are correct
    2. (i) and (ii) are correct
    3. (ii) and (iii) correct
    4. (i) and (iii) correct
22. Compared to axial compressors, centrifugal compressors are more suitable for
    1. high head, low flow rate
    2. low head, low flow rate
    3. low head, high flow rate
    4. high head, high flow rate
23. Stalling of blades in axial-flow compressor is the phenomenon of
    1. air stream blocking the passage
    2. motion of air at sonic velocity
    3. unsteady, periodic and reversed flow
    4. air stream not able to follow the blade contour
24. In an axial flow compressor:

    α_1 = exit angle from stator, β_1 = inlet angle to rotor, α_2 = inlet angle to stator, and β_2 = outlet angle from rotor.

    The condition to have a 50% degree of reaction is

    1. $\alpha_1 = \beta_2$
    2. $\alpha_2 = \beta_1$

3. $\alpha_1 = \beta_2$ and $\beta_1 = \alpha_2$
4. $\alpha_1 = \alpha_2$ and $B_1 = \beta_2$

25. Consider the following statements with reference to supercharging of IC engines:
    1. Reciprocating compressors are invariably used for high degree of super charging
    2. Rotary compressors like roots blowers are quite suitable for low degree of supercharging
    3. Axial flow compressors are most commonly employed for supercharging diesel engines used in heavy duty transport vehicles
    4. Centrifugal compressors are used for turbo-charging

    Which of the statements given above are correct?

    1. (i) and (ii)
    2. (ii) and (iii)
    3. (i) and (iv)
    4. (ii) and (iv)
26. While flowing through the rotor blades in an axial flow air compressor, the relative velocity of air
    1. continuously decreases
    2. continuously increases
    3. first increases and then decreases
    4. first decreases and then increases
27. In an axial flow compressor, when the degree of reaction is 50%, it implies that
    1. work done in compression will be the least
    2. 50% stages of the compressor will be ineffective
    3. pressure after compression will be optimum
    4. the compressor will have symmetrical blades
28. For a multistage compressor, the polytropic efficiency is
    1. the efficiency of all stages combined together.
    2. the efficiency of one stage.
    3. constant throughout for all the stages
    4. a direct consequence of the pressure ratio

Explanatory Notes

1. 16. (d) Degree of reaction,

Review Questions

1. How do axial flow compressors differ from centrifugal compressors?
2. Explain the working principle of axial flow compressors.
3. Define lift and drag coefficients.
4. Define degree of reaction and write down the mathematical expression for it.
5. For 50% degree of reaction, what is the condition on blade angles?
6. Write the expression for pressure rise in isentropic flow through a cascade.
7. Define polytropic efficiency and write down the mathematical expression for it.
8. Define flow coefficient and work coefficient.
9. Define deflection coefficient and pressure coefficient.
10. Differentiate between choking and stalling.
11. List the various losses in an axial flow compressors.
12. List the main applications of axial flow compressors.
13. List the merits and demerits of axial flow compressors.
14. Draw the efficiency and power consumption curves for the axial flow compressor.

Exercises

15.1 An axial flow air compressor of 50% degree of reaction has blades with inlet and outlet blade angles of 40° and 15°, respectively. The pressure ratio is 5 : 1 with an overall isentropic efficiency of 85% when the air inlet temperature is 30°C. The blade speed and axial

velocity are constant throughout the compressor. Assuming a blade speed of 210 m/s, find the number of stages required when the work coefficient is 0.9 for all stages.

15.2 In an axial flow air compressor, the overall stagnation pressure ratio is 4.5 with overall stagnation isentropic efficiency 85%. The inlet stagnation pressure and temperature are 1 bar and 310 K. The mean blade speed is 180 m/s and degree of reaction is 0.5 at the mean radius. The relative air angles are 10° and 30° at rotor inlet and outlet, respectively. The work coefficient is 0.9. Calculate (a) the stagnation polytropic efficiency, (b) the number of stages, (c) the inlet temperature and pressure, and

(d) the blade height in the first stage if the hub-tip ratio is 0.45 and mass flow rate is 20 kg/s.

15.3 A multi-stage axial flow compressor delivers 19 kg/s of air. The inlet stagnation condition is 1 bar and 25°C. The power consumed by the compressor is 4400 kW. Calculate (a) the delivery pressure, (b) the number of stages, and (c) the overall isentropic efficiency of the compressor.

Assume that temperature rise in the first stage is 21°C. The polytropic efficiency of compressor is 90% and stagnation pressure ratio is constant.

15.4 An axial flow compressor compresses the air up to overall

stagnation pressure of 10 bar with overall stagnation isentropic efficiency of 88%. The inlet stagnation pressure and temperature are 1 bar and 300 K. The mean blade speed is 200 m/s. The degree of reaction is 0.5 at the mean radius with air angles of 30° and 10° at rotor inlet and outlet with axial direction respectively. The work done factor is 0.88. The hub-tip ratio is 0.4. The mass flow rate is 50 kg/s. Draw the inlet and outlet velocity triangles and show the compression process on *T-s* diagram. Find the following:

1. The stagnation polytropic efficiency
2. The number of stages
3. The blade height in the first stage of the compressor

15.5 An axial flow compressor provides a total head pressure ratio of 4 : 1 with an overall total head isentropic

efficiency of 85%, when the inlet total head temperature is 290 K. The compressor is designed for 50% reaction with inlet and outlet air angles from the rotor blades of 45° and 10°, respectively. The mean blade speed and axial velocity are constant throughout the compressor. Assuming a value of 201.16 m/s for the blade speed and a work done factor of 0.86, find the number of stages required. What is the inlet Mach number relative to the rotor at the mean blade height of the first stage?

15.6 Air at a temperature of 300 K enters a 10-stage axial flow compressor at the rate of 3.5 kg/s. The pressure ratio is 6.0 and the isentropic efficiency is

90%. The compression process is adiabatic and the compressor has symmetrical stages. The axial velocity is uniform across the stages and equals to 120 m/s and the mean blade speed is 200 m/s. Assume that the temperature change is same in each stage. Determine the direction of the air at entry and exit from the rotor and stator blades. Also, find the power given to the air. For air, take c_p = 1.005 kJ/kg.K and γ = 1.4.

15.7 An axial flow compressor consists of a number of similar stages with equal work done per stage. The velocity of flow is uniform throughout the compressor. The data given is as follows:

Calculate (a) the stagnation outlet temperature, and (b) the number of stages.

ANSWERS TO MULTIPLE-CHOICE QUESTIONS

1. b
2. a
3. c
4. c
5. a
6. a
7. b
8. b
9. c
10. c
11. b
12. c
13. a
14. c
15. c
16. d
17. a
18. b
19. c
20. a
21. d
22. a
23. c
24. c
25. d
26. a
27. d
28. a

Chapter 16

Gas Turbines

16.1 ❐ INTRODUCTION

The gas turbine is the most satisfactory way of producing very large power in a self-contained and compact unit. The gas turbine obtains its power by utilising the energy of a jet or burnt gases and air, the velocity of the jet being absorbed as it flows over several rings of moving blades which are fixed to a common shaft. The gas turbine requires an air compressor which is driven off its own shafting. This absorbs a considerable proportion of the power produced and thus lowers the overall efficiency, which

is of the order of 20%–30%.

16.2 ❐ FIELDS OF APPLICATION OF GAS TURBINE

The following are the major fields of application of gas turbines:

1. **Aviation:** A gas turbine is used in aviation because it is self-contained, light weight not requiring cooling water, and generally fits into the overall shape of the aircraft structure. It is also suitable for turbo-jet and turbo-propeller engines.
2. **Power generation:** It is used for electric power generation because of its simplicity, lack of cooling water, quick installation, and starting. It is also suitable for supercharging, road transport, and railways.
3. **Oil and gas industry:** It is used in the oil and gas industry due to cheaper supply of fuel and low installation cost.
4. **Marine propulsion:** It is used in marine propulsion due to its light weight and it does not require cool water.

16.3 ❐ LIMITATIONS OF GAS TURBINES

The limitations of gas turbines are as follows:

1. They are not self-starting.
2. Low efficiency at part loads
3. Higher rotor speeds
4. Low overall plant efficiency

16.4 ❐ COMPARISON OF GAS TURBINES WITH IC ENGINES

16.4.1 Advantages

1. Perfect balancing of the rotating parts is possible.
2. Mechanical efficiency is very high (≈ 95%) compared with 85% of IC engine because of its large number of sliding parts.
3. Torque on the turbine shaft is continuous and uniform; therefore, flywheel is not required.
4. Work developed is more because expansion of gases up to atmospheric pressure is possible.
5. The weight of the gas turbine per kW of power developed is less, and therefore, they are superior for use in aircraft.
6. The compression, combustion, and expansion take place in different units. Therefore, these units can be designed, tested, and developed individually.
7. The gas turbine can be driven at much higher speeds.
8. The maximum pressure used in gas turbines is very low; therefore, its components can be made lighter.
9. The lubrication and ignition systems are much simpler.
10. Cheaper fuels can be used in gas turbine.
11. The installation and maintenance costs are less.
12. The exhaust is free from smoke and is less polluting.

16.4.2 Disadvantages

1. The thermal efficiency of a simple gas turbine is low.
2. There is very poor thermal efficiency at part loads.
3. The temperature of gases supplied to the turbine is limited to 1100 K as there are no metals which can be used for blades to sustain this temperature at high centrifugal stresses.
4. The manufacturing of blades is difficult and costly as Ni-Cr alloy is used.
5. The control of fuel is difficult with wide operating speeds and output.
6. A special cooling system is required for the blades.
7. The starting is more difficult than IC engines.
8. It produces five times exhaust gases than an IC engine for same output.
9. Speed reduction device is always required due to higher operating speed.
10. The life of blades and combustion chamber is low due to high temperature.

16.5 ❐ ADVANTAGES OF GAS TURBINES OVER STEAM TURBINES

1. Boiler, along with accessories, is not required.
2. The capital and running cost is less.
3. Less space is required for the plant.
4. Starting the plant is more easy and quick.
5. The weight of the plant per kW generated is less.
6. Control of plant is much easier.
7. The plant can be installed anywhere as cooling water is not required.
8. Gas turbine plant is used as a best type for peak load plant.

16.6 ❐ CLASSIFICATION OF GAS TURBINES

Gas turbines may be classified as follows:

1. Constant pressure combustion (or continuous combustion) gas turbine
    1. Open cycle constant pressure gas turbine
    2. Closed cycle constant pressure gas turbine
2. Constant volume combustion (or explosion) gas turbine

16.6.1 Constant Pressure Combustion Gas Turbine

In this type, the fuel is burned at constant pressure and the cycle used is the Joule cycle. The turbine belongs to the reaction type using oil fuel and is fitted with an axial flow rotary air

compressor which is coupled to the rotor shaft. The heat is supplied to the working fluid at constant pressure.

The open and closed cycle gas turbines are shown in Figs 16.1(a) and 16.1(b), respectively. A simple open cycle constant pressure gas turbine consists of a compressor, combustion chamber, and turbine. The air is taken from the atmosphere by the compressor and is compressed to the combustion pressure of 1 to 4 atm. It is then forced into the combustion chamber. A part of this air is used as combustion air for the oil which enters the burner, and the remainder is forced through the annular space between the wall of the combustion chamber and the burner

jacket. The air receives heat from the burner jacket and also mixes with the products of combustion chamber and the burner jacket. This raises the temperature and volume of the air. The use of a very large quantity of air in excess of the combustion air prevents the temperature of the mixture from reaching values which are too high for the metal of the rotor blades. It also prevents the burner from becoming too hot.

The high pressure and high temperature gases now enter the turbine and flow through the blade rings, where they continuously expand, and the pressure energy is converted into kinetic energy, which in turn is absorbed by the rotor.

On leaving the turbine, the spent gases pass away to the exhaust. The speed of the turbine is generated by controlling the fuel oil supply and the safety valve which by-passes some of the mixture. If the pressure becomes too high, the net power developed by the turbine is utilised in driving the generator but a considerable amount of the total power produced by the rotor is absorbed in driving the compressor. The turbine is started by the electric starter motor.

In the closed cycle constant pressure gas turbine, the compressed fluid coming out from the compressor is heated in the heater by an external source at constant pressure and the high pressure and temperature fluid develop the work

while passing through the turbine. The fluid coming out from the turbine is cooled to its original temperature in the cooler using an external cooling source before passing into the compressor. Thus, the working fluid is continuously circulated in the closed cycle. Such a turbine has now become obsolete.

Figure 16.1 *Constant pressure combustion gas turbine: (a) Open cycle turbine, (b) Closed cycle turbine*

16.6.2 Constant Volume Combustion Gas Turbine

The constant volume combustion gas turbine is shown in Fig. 16.2. Air is taken from the atmosphere and is compressed by an axial flow compressor driven by a separate steam turbine. The compressed air coming out of the compressor at about 3 bar is passed to the constant volume combustion

chamber. The fuel is injected into the combustion chamber in the current of air by a separate fuel pump. The air-fuel mixture formed in the combustion chamber is then ignited by means of a spark plug. The combustion takes place at constant volume in the combustion chamber and the pressure inside the chamber increases to nearly 12 bar. The high pressure and temperature gases are fed to the gas turbine through the water-cooled nozzle. The gases passed through the turbine develop the useful work. The exhaust gases are fed to the steam boiler to generate steam for the steam turbine driving the compressor.

Figure 16.2 *Constant volume combustion gas turbine*

The major disadvantage of this turbine

is the intermittent combustion which impairs its smooth functioning. The absolute necessity of a separate steam turbine to recover the heat from exhaust gases further discourages its use in practice.

16.7 ❐ COMPARISON OF OPEN AND CLOSED CYCLE GAS TURBINES

The comparison of open and closed cycles gas turbines is given in Table 16.1.

Table 16.1 *Comparison of open and closed cycles gas turbines*

16.8 ❐ POSITION OF GAS TURBINE IN THE POWER INDUSTRY

Constant pressure gas turbines are now used for generating electricity, driving locomotives and aeroplane propellers,

producing the air stream for wind tunnels, and jet propulsion of air craft. In some types of aircraft, the exhaust gases from the main engines are used for driving a gas turbine which, in turn, drives the supercharger. Gas turbines driven by exhaust gases are also used for supplying a pressure charge to diesel engines.

Figure 16.3 *Gas turbine utilising exhaust heat*

Overall efficiencies of constant pressure gas turbine plants vary between 18%–25%. This is based on a turbine efficiency of 75%–78%. The efficiency of this type of turbine can be improved by using heat from exhaust gases to preheat the compressed air before it enters the combustion chamber. The hot

mixture of air and burnt gases is then expanded through the turbine from which they are exhausted into the heat exchanger. The available heat is now given to the high pressure air from the compressor. Such a system is shown in Fig. 16.3.

16.9 ❐ THERMODYNAMICS OF CONSTANT PRESSURE GAS TURBINE: BRAYTON CYCLE

16.9.1 Efficiency

The Brayton cycle is shown on *p-V* and *T-s* diagrams in Fig. 16.4. During this cycle, air is drawn into the compressor at point 1 and then compressed isentropically along the process 1-2. Process 2-3 represents the burning of the oil at constant pressure p_2. Process 3-4 is the isentropic expansion of the gases through the turbine. Process 4-1

represents the exhausting and cooling of the gases at constant pressure p_1.

For 1 kg of air,

Compressor work, $w_c = w_{1-2} = h_2 - h_1 = c_p (T_2 - T_1)$

Heat supplied, $q = q_{2-3} = h_3 - \grave{}h_2 = c_p (T_3 - T_2)$

Turbine work, $w_t = w_{3-4} = h_3 - h_4 = c_p (T_3 - T_4)$

Net work done, $w_{\text{net}} = w_t - w_c = c_p [(T_3 - T_4) - (T_2 - T_1)]$

Figure 16.4 *Brayton cycle for constant pressure gas turbine: (a) p-V diagrams, (b) T-s diagrams*

Let be the pressure ratio and

For the adiabatic compression process 1-2, we have

and for the adiabatic expansion process 3-4, we have

But $p_1 = p_4$ and $p_2 = p_3$

∴

Thermal efficiency,

Figure 16.5 *Variation of thermal efficiency with pressure ratio*

The variation of thermal efficiency with r_p is shown in Fig. 16.5.

16.9.2 Specific Output

Specific output

For maximum specific output,

or

Figure 16.6 *Variation of specific output with pressure ratio*

Hence, for maximum work output, the temperature after compression must be equal to the exhaust gas temperature, which depends on the expansion ratio. The variation of specific output with r_p is shown in Fig. 16.6.

16.9.3 Maximum Work Output

Maximum work output,

$$(w_{net})_{max} = c_p [(T_3 - T_4) - (T_2 - T_1)]$$
$$= c_p [T_3 + T_1 - T_4 - T_2]$$

16.9.4 Work Ratio

16.9.5 Optimum Pressure Ratio for Maximum Specific Work Output

$$w_{\text{net}} = w_t - w_c = c_p\,(T_3 - T_4) - c_p\,(T_2 - T_1)$$

w_{net} will be maximum, when

Let

From Eq. (16.3), we have

Combining with Eq. (16.7), we get

16.10 ❒ CYCLE OPERATION WITH MACHINE EFFICIENCY

16.10.1 Maximum Pressure Ratio for Maximum Specific Work

It is not possible to achieve isentropic compression and expansion in the actual

Brayton cycle because of inevitable losses due to friction and turbulence in the compressor and turbine. Therefore, the temperature at the end of compression and expansion are higher than that in an ideal cycle for the same pressure ratio. The actual Brayton cycle is shown in Fig. 16.7.

Isentropic efficiency of compressor,

Isentropic efficiency of turbine,

Net specific work available,

$$w_{\text{net}} = w_t - w_c = c_p\,(T_3 - T_{4'}) - c_p\,(T_{2'} - T_1)$$

For specific work output to be maximum for given temperature limits,

Figure 16.7 *Actual Brayton cycle with machine efficiencies*

16.10.2 Optimum Pressure Ratio for Maximum Cycle Thermal Efficiency

Heat supplied, $q = c_p (T_3 - T_{2'}) = c_p (T_3 - T_2)$

The thermal efficiency of the cycle is dependent only on r_p for the fixed values of T_1 and T_3. The condition for the maximum value of thermal efficiency for given temperature limits is

Let

Taking -ve sign

16.11 ❐ OPEN CYCLE CONSTANT PRESSURE GAS TURBINE

The open cycle constant pressure gas turbine plant is shown diagrammatically in Fig. 16.8. The fuel is burned with air coming out from the compressor into the combustion chamber. The products of combustion from the combustion chamber are passed through the turbine and developed the required power. Part of the power developed is used to run the compressor. The gases leaving the turbine are exhausted to atmosphere.

Let $\dot{m}_a$ = mass rate of flow of air

$\dot{m}_f$ = mass rate of flow of fuel

Compressor work, $W_c = \dot{m}_a c_{pa} (T_2 - T_1)$

Turbine work, $W_t = (\dot{m}_a + \dot{m}_f) c_{pg} (T_3 - T_{4'})$

where c_{pa}, c_{pg} = specific heat of air and gases, respectively

Network $W_{net} = W_t - W_c = (\dot{m}_a + \dot{m}_f) c_{pg} (T_3 - T_{4'}) - \dot{m}_a c_{pa} (T_{2'} - T_1)$

Heat supplied, $Q = \dot{m}_f \times$ C.V.

where CV = calorific value of fuel

Figure 16.8 *Open cycle constant pressure gas turbine: (a) Schematic diagram, (b) T-s diagrams*

Thermal efficiency,

16.12 ❐ METHODS FOR IMPROVEMENT OF THERMAL EFFICIENCY OF OPEN CYCLE CONSTANT PRESSURE GAS TURBINE

The thermal efficiency of open cycle constant pressure gas turbine can be improved by the following methods:

1. Regeneration
2. Intercooling
3. Reheating
4. Combination of above

16.12.1 Regeneration

In this method, the heat of the exhaust gases is used to heat the air coming out from the compressor, thus reducing the mass of fuel supplied in the combustion chamber. A schematic diagram of such a plant is shown in Fig. 16.9 (a).

1. **Without Machine Efficiencies**

   The *T-s* diagram without machine efficiencies is shown in Fig. 16.9 (b).

   For 1 kg of air

   Compressor work, $w_c = w_{1-2} = c_p (T_2 - T_1)$

Heat supplied, $q_s = q_{5-3} = c_p (T_3 - T_5)$

Turbine work, $w_t = w_{3-4} = c_p (T_3 - T_4)$

Net work, $w_{net} = w_t - w_c = c_p (T_3 - T_4) - c_p (T_2 - T_1)$

Figure 16.9 *Open cycle gas turbine with regeneration: (a) Schematic diagram, (b) T-s diagram without machine efficiencies, (c) T-s diagram with machine efficiencies*

Thermal efficiency,

In an ideal regenerator, $T_4 = T_5$

Figure 16.10 *Variation of thermal efficiency with pressure ratio for open cycle gas turbine with regeneration*

The variation of thermal efficiency with pressure ratio is shown in Fig. 16.10. It may be seen that thermal efficiency increases with an increase in and decreases with increase in r_p.

2. **Considering Machine Efficiencies**

The *T-s* diagram is shown in Fig. 16.9 (c).

Figure 16.11 *Actual regeneration process*

3. **Effectiveness of Regenerator**

In actual practice, the rise in temperature from $T_{2'}$ to $T_5 = T_{4'}$ is not possible. The actual temperature of air will be $T_{5'} < T_5$. Hence, actual thermal efficiency [see Fig. (16.11)] of the cycle

becomes,

The effectiveness of the regeneration is given by

where $\dot{m}_a$, $\dot{m}_g$ = mass rate of flow of air and exhaust gases, respectively

c_{pa}, c_{pg} = specific heats of air and exhaust gases, respectively.

16.12.2 Intercooling

The work consumed by the compressor can be reduced by compressing the air in two stages and incorporating the intercooler in between, as shown in Fig. 16.12 (a). The corresponding *T-s* diagram is shown in Fig. 16.12 (b).

Figure 16.12 *Gas turbine plant with intercooling: (a) Schematic diagram, (b) T-s diagram*

For perfect cooling, $T_3 = T_1$ and if $n_{c1} = n_{c2} = n_{c'}$ then

$$w_{net} = wt - w_c = c_p (T_5 - T_{6'}) - [c_p (T_{2'} - T_1) + c_p (T_{4'} - T_3)]$$

For perfect intercooling,$T_3 = T_1$ and if h$_{c1} = \eta_{c2} = \eta_c$, then

16.12.3 Reheating

A considerable increase in power output can be achieved by expanding the gases in two stages with a reheat combustion chamber between the two as shown in Fig. 16.13 (a). The corresponding *T-s* diagram is shown in Fig. 16.13 (b).

$$w_t = c_{pg}\,(T_3 - T_4') + c_{pg}\,(T_5 - T_6')$$
$$= c_{pg}\,\eta_{t1}\,(T_3 - T_4) + c_{pg}\,\eta_{t2}\,(T_5 - T_6)$$

where

Let

Figure 16.13 *Gas turbine plant with reheating: (a) Schematic diagram, (b) T-s diagram*

If $\eta_{t1} = \eta_{t2} = \eta_t$ and $T_5 = T_3$, then

16.12.4 Reheat and Regenerative Cycle

This cycle is shown in Fig. 16.14.

$w_c = c_p\,(T_2 - T_1)$

$q_s = q_{7-3} + q_{4-5} = c_p\,(T_3 - T_7) + c_p\,(T_5 - T_4)$

$w_t = w_{3-4} + w_{5-6} = c_p (T_3 - T_4) + c_p (T_5 - T_6)$

Figure 16.14 *Gas turbine plant with reheat and regeneration: (a) Schematic diagram, (b) T-s diagram*

Net work $w_{\text{net}} = w_t - w_c$

$$= c_p [(T_3 - T_4) + (T_5 - T_6) - (T_2 - T_1)]$$

Let so that $r_p = r_1 r_2$

So that $c = c_1 c_2$

Now,

Now, $T_7 = T_4$ and $T_5 = T_3$

16.12.5 Cycle with Intercooling and Regeneration

The schematic arrangement and *T-s* diagram of the cycle are shown in Fig. 16.15.

$w_c = w_{1-2} + w_{3-4} = c_p [(T_2 - T_1) + (T_4 - T_3)]$

$q_s = q_{7-5} = c_p (T_5 - T_7)$

$w_t = w_{5-6} = c_p (T_5 - T_6)$

$w_{\text{net}} = w_t - w_c = c_p [(T_5 - T_6) - (T_2 - T_1) - (T_4 - T_3)]$

Let so that $c = c_1 c_2$ and $T_3 = T_1$

Figure 16.15 *Gas turbine plant with inter cooler and regeneration: (a) Schematic diagram, (b) T-s diagram*

Now, $q_s = c_p (T_5 - T_7)$

16.12.6 Cycle with Intercooling and Reheating

The schematic arrangement of this cycle is shown in Fig. 16.16 (a) and the corresponding *T-s* diagram in Fig. 16.16 (b).

$$w_c = c_p\,[(T_2 - T_1) + (T_4 - T_3)]$$

$$q_s = c_p\,[(T_5 - T_4) + (T_7 - T_6)]$$

$$w_t = c_p\,[(T_5 - T_6) + (T_7 - T_8)]$$

$$w_{\text{net}} = w_t - w_c = c_p\,[(T_5 - T_6) + (T_7 - T_8) - (T_2 - T_1) - (T_4 - T_3)]$$

Now, $T_3 = T_1$ and $T_7 = T_5$ for perfect intercooling and reheating, respectively.

Then,

Thus, $c = c_1 c_2$

For maximum work output,

or

Figure 16.16 *Gas turbine cycle with intercooler and reheating: (a) Schematic diagram, (b) T-s diagram*

16.12.7 Cycle with Intercooling, Regeneration and Reheating

The schematic arrangement of the cycle and corresponding *T-s* diagram are shown Fig. 16.17.

$w_c = c_p\,[(T_2 - T_1) + (T_4 - T_3)]$

$$q_s = c_p\,[(T_5 - T_9) + (T_7 - T_6)]$$

$$w_t = c_p\,[(T_5 - T_6) + (T_7 - T_8)]$$

$$w_{\text{net}} = w_t - w_c = c_p\,[(T_5 - T_6 + T_7 - T_8) - (T_2 - T_1 + T_4 - T_3)]$$

$$T_7 = T_5$$

Let $T_1 = T_3$ and $T_5 = T_7$ so that $c_1 = c_2$

Figure 16.17 *Gas turbine cycle with intercooling, regeneration and reheating: (a) Schematic diagram, (b) T-s diagram*

Similar to the case of 16.12.6, it can be shown that

Example 16.1

Calculate the indicated mean effective pressure and efficiency of a Joule cycle if the temperature at the end of combustion is 2000 K and the temperature and pressure before compression are 350 K and 1 bar, respectively. The pressure ratio is 1.3 and assume $c_p = 1.005$ kJ/kgK.

Solution

Refer to Fig. 16.18.

$p_1 = 1$ bar, $T_1 = 350$ K, $T_3 = 2000$ K

$r_p = 1.3$, $c_p = 1.005$ kJ/kgK

$w_c = c_p (T_2 - T_1) = 1.005 (377.24 - 350) = 27.376$ kJ/kg

$w_t = c_p\,(T_3 - T_4) = 1.005\,(2000 - 1855.56) = 145.162$ kJ/kg

Net work done, $w_{\text{net}} = w_t - w_c = 145.162 - 27.376 = 117.786$ kJ/kg

$q_s = c_p\,(T_3 - T_2) = 1.005\,(2000 - 377.24) = 1630.874$ kJ/kg

Thermal efficiency

Figure 16.18 *p-v diagram*

Example 16.2

A gas turbine operates on a pressure ratio of 6. The inlet air temperature to the compressor is 300 K and the

temperature of air entering to the turbine is 580°C. If the volume rate of air entering the compressor is 250 m³/s, calculate the net power output of the cycle. Also, calculate the efficiency.

Solution

The *T*-*s* diagram shown in Fig. 16.19

$$r_p = 6,\ T_1 = 300\ \text{K},\ T_3 = 273 + 580 = 853\ \text{K}$$

$$\dot{V}_1 = 250\ \text{m}^3/\text{s}$$

Compressor work $w_c = c_p\,(T_2 - T_1)$ = 1.005 (500.55 – 300) = 201.55 kJ/

kg

Turbine work $w_t = c_p\ (T_3 - T_4) =$ 1.005 (853 – 511.23) = 343.48 kJ/kg

Net work output $w_{\text{net}} = w_t - w_c =$ 343.48 – 201.55 = 141.93 kJ/kg

Heat input $q_s = c_p\ (T_3 - T_2) =$ 1.005 (853 – 500.55) = 354.21 kJ/kg

Thermal efficiency

Figure 16.19 *T-s diagram*

Example 16.3

A gas turbine has a perfect heat exchanger. Air enters the compressor at a temperature and pressure of 300 K and 1 bar and discharges at 475 K and 5 bar. After passing through the heat exchanger, the air temperature increases to 655 K. The temperatures of air entering and leaving the turbine are 870°C and 450°C, respectively. Assuming no pressure drop through the heat exchanger, compute (a) the output per kg of air, (b) the efficiency of cycle, and (c) work required to drive the compressor.

Solution

The p – V and T-s diagrams are shown in Fig. 16.20.

$T_1 = 300$ K, $p_1 = 1$ bar,

$T_2 = 475$ K, $p_2 = 5$ bar,

$T_5 = 655$ K, $T_3 = 870 + 273 = 1143$ K,

$T_4 = 450 + 273 = 723$ K

or $\gamma = 1.4$, $c_p = 1.005$ kJ/kg

Compressor work $w_c = c_p (T_2 - T_1) = 1.005 (475 - 300) = 175.9$ kJ/kg

Turbine work $w_t = c_p (T_3 - T_4) = 1.005 (1143 - 723) = 422.1$ kJ/kg

Heat input $q_s = c_p (T_3 - T_5) = 1.005 (1143 - 655) = 490.44$ kJ/kg

1. Net work output, $w_{\text{net}} = w_t - w_c = 422.1 - 175.9 = 246.2$ kJ/

kg

2. Thermal efficiency,
3. Compressor work, w_c = 175.9 kJ/kg

Figure 16.20 *Diagrams for a gas turbine: (a) p-V diagram, (b) T-s diagram*

Example 16.4

A closed cycle regenerative gas turbine operating with air has the following data: p_1 = 1.4 bar, T_1 = 310 K, r_p = 5, T_{max} = 1050 K, effectiveness of generator = 100%, net output = 3 MW. Calculate (a) the mass flow rate of air per minute and (b) the thermal efficiency.

Solution

The cycle in *T-s* diagram is shown in Fig. 16.21.

Refer to Fig. 16.21

Figure 16.21 *T-s diagram for closed cycle regenerative gas turbine*

For 100% regenerative effectiveness,

$$T_4 = T_5 = 662.65 \text{ K}$$

1. $w_{\text{net}} = \dot{m}c_p\,(T_3 - T_4\,) - \dot{m}c_p\,(T_2 - T_1)$

$$3000 = \dot{m} \times 1.005\ [(1050 - 662.65) - (491.2 - 310)]$$

$$\dot{m} = 14.48 \text{ kg/s or } 868.8 \text{ kg/min}$$

2. Heat input $Q = \dot{m}c_p\,(T_3 - T_5)$

$$= 14.48 \times 1.005\ (1050 - 662.65) = 5636.87 \text{ kJ/s}$$

Thermal efficiency

Example 16.5

The following data refers to a turbine with regenerator:

$T_1 = 290$ K, $T_2 = 460$ K, $T_3 = 900°C$, $T_4 = 467°C$

Calculate (a) the pressure ratio, (b) the specific work output, (c) the efficiency of cycle, and (d) the compressor work. Assume $\eta_{mech} = 100\%$.

Figure 16.22 *T-s diagram for a gas turbine with regenerator*

Solution

The *T-s* diagram of the cycle is shown in Fig. 16.22.

1.

or $r_p = 5$

2. Work output, $w_{net} = c_p\,[(T_3 - T_4) - (T_2 - T_1)]$
$= 1.005[(1173 - 740) - (460 - 290)]$
$= 264.3$ kJ/kg
3. Heat input, $q_s = c_{th}\,(T_3 - T_5) = c_p\,(T_3 - T_4)$
$= 1.005\,(1173 - 740) = 435.165$ kJ/kg

Thermal efficiency

4. Compressor work, $w_c = c_p\,(T_2 - T_1) = 1.005\,(460 - 290) =$ 170.85 kJ/kg

Example 16.6

The ratio of net work to turbine work of an ideal gas turbine plant is 0.563. The temperature of air at the inlet to the compressor is 300 K. Calculate the temperature drop across the turbine if thermal efficiency of the unit is 35%. Assume a mass flow rate of 10 kg/s, $c_p = 1$ kJ/kgK and $\gamma = 1.4$.

Solution

Thermal efficiency

or

or $r_p = 4.51$

Now

Compressor work $w_c = c_p \, (T_2 - T_1)$
$= 1 \times (461.54 - 300) = 161.54$ kJ/kg

or $w_t = 369.66$ kJ/kg $= c_p \cdot \Delta T$

$$\Delta T = 369.66°\text{C}$$

Example 16.7

In a Brayton cycle, the pressure

ratio in the compressor is r_p. The minimum and maximum temperatures are T_1 and T_3. The air is expanded in two stages, each turbine having the same pressure ratio, with reheat to T_3 between the stages. Show that the specific work output will be maximum when

Solution

The *T-s* diagram is shown in Fig. 16.23.

Refer to Fig. 16.23.

Turbine work $w_t = c_p\,(T_3 - T_4) + c_p\,(T_5 - T_6)$

Figure 16.23 *T-s diagram for Brayton cycle with two-stage turbine*

$$= c_p (T_3 - T_4) + c_p (T_3 - T_6)$$

Compressor work

$$= c_p T_1 (c - 1)$$

Specific work output

For maximum specific work output

Example 16.8

In a gas turbine, air is taken in at 1 bar pressure and 30°C. It is compressed to 6 bar with an isentropic efficiency of 87%. Heat is added by the combustion of fuel in combustion chamber to raise the

temperature to 700°C. The efficiency of the turbine is 85%. The calorific value of fuel used is 43.1 MJ/kg. For an air flow of 80 kg/min, calculate (a) the air/fuel ratio of turbine gases, (b) the final temperature of exhaust gases, (c) the net power developed, and (d) the overall thermal efficiency of plant.

Assume c_{pa} = 1.005 kJ/kgK, γ_a = 1.4, c_{pg} = 1.147 kJ/kgK, γ_g = 1.33.

Figure 16.24 *T-s diagram for gas turbine with machine efficiencies*

Solution

The *T-s* diagram is shown in Fig. 16.24.

p_1 = 1 bar, T_1 = 273 + 30 = 303 K

$p_2 = 6$ bar, $\eta_c = 0.87$, $T_3 = 273 + 700$
$= 973$ K

$\eta_t = 0.85$, CV = 43.1 MJ/kg

1. $\dot{m}_f \times \text{c.v} = \dot{m}_a c_{pg} (T_3 - T_{2'})$ $\quad [\because \dot{m}_a + \dot{m}_f \simeq \dot{m}_a]$
2.
3. $w_{net} = \dot{m}_a c_{pg} (T_3 - T_{4'}) - \dot{m}_a c_{pa} (T_{2'} - T_1)$
4.

Example 16.9

In a gas turbine plant, air enters the compressor at 1 bar and 7°C. It is compressed to 4 bar with an isentropic efficiency of 82%. The maximum temperature at the inlet to the turbine is 800°C. The isentropic efficiency of turbine is 85%. The calorific value of fuel used is 43100 kJ/kg. The heat losses are 15% of

the calorific value. Calculate (a) the compressor work, (b) the heat supplied, (c) the turbine work, (d) the net work, (e) the thermal efficiency, (f) the air-fuel ratio, (g) the specific fuel consumption, and (h) the ratio of compressor to turbine work. Assume $c_{pa} = 1.005$ kJ/kgK, $\gamma_a = 1.4$, $c_{pg} = 1.147$ kJ/kgK, $\gamma_g = 1.33$.

Solution

$p_1 = 1$ bar, $T_1 = 280$ K, $\eta_c = 0.82$, $p_2 = 4$ bar, $T_3 = 1073$ K, $\eta_t = 0.85$, $\eta_{\text{comb}} = 0.85$.

The *T-s* diagram is shown in Fig. 16.25.

Refer to Fig 16.25

1. The compressor work

$$w_c = c_{pa} (T_{2'} - T_1) = 1.005 (446 - 280) = 166.82 \text{ kJ/kg}$$

Figure 16.25 *T-s diagram for gas turbine with machine efficiencies*

2. The heat supplied

3.

$$T_{4'} = T_3 - \eta_t (T_3 - T_4) = 1073 - 0.85 (1073 - 760.7) = 807.5 \text{ K}$$

The turbine work

$$w_t = c_{pg} (T_3 - T_{4'}) = 1.147 (1073 - 807.5) = 304.53 \text{ kJ/kg}$$

4. The net work output

$$w_{\text{net}} = w_t - w_c = 304.153 - 166.83 = 137.7 \text{ kJ/kg}$$

5. The thermal efficiency
6. $\dot{m_f} \times$ C.V. (1 - 0.15) = $(\dot{m_a} + \dot{m_f})\ Q \times \eta_{comb}$
7. The specific fuel consumption
8.

Example 16.10

In a closed circuit gas turbine plant, the working fluid at 38°C is compressed with an adiabatic efficiency of 82%. It is then heated at constant pressure to 650°C. The fluid expands down to initial pressure in a turbine with an adiabatic efficiency of 80%. The pressure ratio is such that work done per kg is maximum. c_{pa} = 1.005 kJ/kgK, c_{pg} = 1.147 kJ/kgK. Calculate the pressure ratio for maximum work output and the corresponding cycle efficiency.

Solution

Refer to Fig 16.25.

For maximum specific work,

Example 16.11

A constant pressure closed cycle turbine plant works between a temperature range of 800°C and 30°C. The isentropic efficiencies of compressor and turbine are 80% and 90%, respectively. Find the pressure ratio of the cycle for maximum thermal efficiency and maximum specific output of the cycle.

Solution

$$T_1 = 273 + 30 = 303 \text{ K}, T_3 = 273 + 800 = 1073 \text{ K}, \eta_c = 0.8, \eta_t = 0.9$$

Pressure ratio for maximum specific output

Pressure ratio for maximum thermal efficiency

Example 16.12

A gas turbine plant operates between 5°C and 839°C. Find the following:

1. Pressure ratio at which cycle efficiency equals to Carnot cycle efficiency
2. Pressure ratio at which maximum work is obtained
3. Efficiency corresponding to maximum work output

Solution

The efficiency of Carnot cycle

For simple gas turbine cycle,

1. or
2. For maximum work output

   or
3.

Example 16.13

A simple open cycle gas turbine has a compressor turbine and a free power turbine. It develops electrical power output of 250 MW. The cycle takes in air at 1 bar and 288 K. The total compressor pressure ratio is

14. The turbine inlet temperature is 1500 K. The isentropic efficiency of compressor and turbine are 0.86 and 0.89, respectively. The mechanical efficiency of each shaft is 0.98. Combustion efficiency is 0.98 while combustor pressure loss is 0.03 bar. The alternator efficiency is 0.98. Take calorific value of fuel equal to 42,000 kJ/kg, c_{pa} = 1.005 kJ/kgK and c_{pg} = 1.15 kJ/kgK. Calculate (a) the air-fuel ratio, (b) the specific work output, (c) the specific fuel consumption, (d) the mass flow rate of air, and (e) the thermal efficiency of the cycle.

[IES 2004]

Solution

$p_1 = 1$ bar, $T_1 = 288$ K, $p_2 = 14$ bar, $p_3 = 14 - 0.03 = 13.97$ bar, $T_3 = 1500$ K, $P = 250$ kW, $\eta_c = 0.86$, $\eta_t = 0.89$, $\eta_{\text{mech}} = 0.98$, $\eta_{\text{comb}} = 0.98$, $\eta_a = 0.98$, CV = 42,000 kJ/kg, $c_{pa} = 1.005$ kJ/kgK, $c_{pg} = 1.15$ kJ/kgK

Figure 16.26 *T-s diagram for gas turbine with pressure loss in combustion chamber*

The T-s is shown in Fig. 16.26.

$$T_2 = 288\,(14)^{0.4/1.4} = 612.15 \text{ K}$$

Compressor efficiency,

or $\quad T_{2'} - T_1 = 376.92$ K

or $\quad T_{2'} = 664.92$ K

Turbine efficiency,

Actual net work done, w_{net} = Turbine work, w_t – Compressor work, w_c

Effective heat supplied

$$q_e = c_{pg} (T_3 - T_{2'})$$
$$= 1.15 (1500 - 664.92) = 960.34 \text{ kJ/kg}$$

Actual heat supplied,

1. Air/fuel ratio,
2. Specific work output = 618.72 kJ/kg of air.
3. Specific fuel consumption =
4. Mass flow rate of air, m_a × Specific work output
   = Electric power developed × Alternator efficiency
5. Thermal efficiency of the cycle

Example 16.14

An open cycle gas turbine employs a regenerative arrangement. The air enters the compressor at 1 bar and

288 K and is compressed to 10 bar with a compression efficiency of 85%. The air is heated in the regenerator and the combustion chamber till its temperature is raised to 1700 K and during the process pressure falls by 0.2 bar. The air is then expanded in the turbine and passes to regenerate which has 75% effectiveness and causes a pressure drop of 0.2 bar. The isentropic efficiency of the turbine is 86%. By sketching the gas turbine system and showing the process on *T*-*s* diagram, calculate the thermal efficiency and the power output if mass flow rate of air is 100 kg/s. Take mechanical and alternator efficiency as 98% each. Take c_{pg} =

1.15 kJ/kgK and c_{pa} = 1.005 kJ/kgK.

[IES 2005]

Figure 16.27 *Open cycle gas turbine: (a) Gas turbine system, (b) T-s diagram*

Solution

The gas turbine system and *T-s* diagram are shown in Fig. 16.27.

Work consumed by the compressor,

$$w_c = c_{pa}(T_{2'} - T_1)$$
$$= 1.005\,(603.3 - 288) = 316.876 \text{ kJ/kg}$$

Work done by the turbine

$$w_t = c_{pg}(T_4 - T_{5'}) = 1.15(1700 - 1040.34)$$
$$= 758.61 \text{ k/kg}$$

Net work done = $w_t - w_c$ = 758.61 − 316.876 = 441.73 kJ/kg

Regenerator effectiveness,

$$T_3 = 0.75(1040.34 - 603.3) + 603.3 = 931.08 \text{ K}$$

Heat supplied = c_{pa} $(T_4 - T_3)$ = 1.005(1700 – 931.08) = 772.76 kJ/kg

Thermal efficiency =

Power developed = 100 × 441.73 × 0.98 × 0.98 = 42423.75 kW

Example 16.15

Air enters the compressor of a gas turbine at 100 kPa, 300 K with a volumetric flow rate of 5 m^3/s. The air is compressed in two stages to 1200 kPa with intercooling to 300 K

between stages at a pressure of 350 kPa. The turbine inlet temperature is 1400 K and the expansion occurs in two stages with reheat to 1340 K between the stages at a pressure of 350 kPa. The compressor and turbine stage efficiencies are 87% and 85%, respectively. Draw the schematic diagram of the cycle and indicate the process on *T*-*s* diagram. Determine (a) the thermal efficiency of the cycles, (b) the back work ratio, (c) the net power developed in kW. Assume effectiveness of the regenerator as 80% and c_p = 1.0045 kJ/kgK for air and gas.

[IES 2006]

Solution

The schematic diagram of the cycle and *T*-*s* diagram are shown in Fig. 16.28.

$p_1 = p_a = 100$ kPa, $T_1 = 300$ K, $\dot{V}_1 = 5$ m^3/s, $p_2 = p_3 = p_7 = p_8 = 350$ kPa,
$p_4 = p_6 = 1200$ kPa, $T_1 = T_3 = 300$ K, $T_6 = 1400$ K, $T_8 = 1340$ K,
$\eta_{c1} = \eta_{c2} = 0.87$, $\eta_{t1} = \eta_{t2}$, $\varepsilon = 0.85$, $c_{pa} = c_{pg}$ 1.0045 kJ/kgK

Compressor 1st stage:

Work done, $w_{c1} = c_{pa}\,(T_{2'} - T_1) = 1.0045\,(448.4 - 300) = 149.07$ kJ/kg

Figure 16.28 *Gas turbine with intercooling, reheating, and regeneration: (a) Gas turbine system, (b) T-s diagram*

Compressor 2nd stage:

Work done, $w_{c2} = c_{pa}\,(T_{4'} - T_3) = 1.0045\,(445.50 - 300) = 146.16$ kJ/

kg

Total compressor work, $w_c = w_{c1} + w_{c2} = 295.23$ kJ/kg

Turbine Ist stage:

or $T_{7'} = 1400 - 0.85\ (1400 - 984.55) = 1046.87$ K

Work done, $w_{t1} = c_{pg}\ (T_6 - T_{7'}) = 1.0045\ (1400 - 1046.87) = 354.72$ kJ/kg

Turbine 2nd stage:

or $T_{9'} = 1340 - 0.85\ (1340 - 936.82) = 997.3$ K

Work done, $w_{t2} = c_{pg} (T_8 - T_{9'}) =$ $1.0045 (1340 - 997.3) = 344.24$ kJ/kg

Total turbine work $w_t = w_{t1} + w_{t2} =$ 698.96 kJ/kg

Net work $w_{\text{net}} = w_t - w_c = 698.96 - 295.23 = 403.73$ kJ/kg

Regenerator effectiveness,

or

or $T_{5'} = 886.94$ K

Heat supplied, $q_s = c_{pg} [(T_6 - T_{5'}) + (T_8 - T_{7'})]$

$$= 1.0045 [(1400 - 886.94) + (1340 - 1046.87)]$$
$$= 809.82 \text{ kJ/kg}$$

1. Thermal efficiency,

2. Work ratio =

3. $p_1 V_1 = \dot{m} R T_1$

or

Power developed = $\dot{m} \times w_{net}$ = 580.72 × 403.73
= 234454 kW

Example 16.16

Air enters the compressor of a gas turbine plant operating on Brayton cycle at 1 bar, 27°C. The pressure ratio in the cycle is 6. If $W_t = 2.5\ W_c$, where W_t and W_c are the turbine and compressor work, respectively, calculate the maximum temperature and the thermal efficiency of the cycle. Take $\gamma = 1.4$.

Solution

p_1 = 1 bar, T_1 = 27 + 273 = 300 K, p_2 = 6 bar, γ = 1.4

Refer to Fig. 16.4, we get

Thermal efficiency =

Example 16.17

In an air-standard regenerative gas turbine cycle, the pressure ratio is 5. Air enters the compressor at 1 bar, 300 K and leaves at 490 K. The maximum temperature in the cycle is 1000 K. If the effectiveness of the regenerator and the isentropic efficiency of the turbine are each 80%, determine the cycle efficiency. Assume $\gamma = 1.4$ for air.

Solution

The cycle on the *T-s* diagram is shown in Fig. 16.29 assuming compressor efficiency to be 100%.

Figure 16.29 *T-s diagram for gas turbine with regeneration*

Turbine efficiency, =

or

or $T_{4'} = 705.11$ K

Regenerative effectiveness =

or

or $T_{5'} = 659.116$ K

$$w_c = c_p (T_2 - T_1) = 1.005 \times (475.14 - 300)$$
$$= 176.01 \text{ kJ/kg}$$
$$w_t = c_p (T_3 -) = 1.005 (1000 - 705.11)$$
$$= 296.36 \text{ kJ/kg}$$

Net work, $w_{net} = w_t - w_c = 296.36 -$

176.01

$$= 120.35 \text{ kJ/kg}$$

Heat supplied, $q_s = c_p\,(T_3 - T_{5'})$

$$= 1.005 \times (1000 - 659.116) = 342.58 \text{ kJ/kg}$$

Thermal efficiency =

16.13 ❒ EFFECTS OF OPERATING VARIABLES

The operating variables for the performance of open cycle gas turbine are as follows:

1. Pressure ratio
2. Compressor inlet temperature
3. Turbine inlet temperature
4. Efficiency of the compressor
5. Efficiency of the turbine

These variables affect the thermal efficiency and the work ratio of the gas turbine.

For the simple Brayton cycle,

Thermal efficiency,

Work ratio,

The thermal efficiency for various values of γ and r_p are given in Table 16.2.

The variation of thermal efficiency with pressure ratio for different values of ratio of specific heats is shown in Fig. 16.30. It may be observed that the thermal efficiency increases as the pressure ratio and the ratio of specific heats increases.

The work ratios for different values of r_p

and T_3/T_1 ratio are given in Table 16.3.

Table 16.2 *Effect of pressure ratio on thermal efficiency*

Figure 16.30 *Variation of thermal efficiency with pressure ratio of a gas turbine*

Table 16.3 *Effect of pressure ratio and temperature ratio on work ratio (γ = 1.4)*

Figure 16.31 *Variation of work ratio with pressure ratio of a gas turbine*

The variation of work ratio with pressure ratio for different values of temperature ratio is shown in Fig. 16.31. It may be observed that the work ratio decreases as the pressure ratio increases. However, the work ratio increases with increase in temperature ratio.

16.13.2 Effect of Efficiencies of Compressor and Turbine on Thermal Efficiency

From Eq. (16.11), we have

where

Figure 16.32 *Variation of thermal efficiency with pressure ratio*

Figure 16.33 *Variation of thermal efficiency with temperature ratio*

Let then

For

The effect of machine efficiencies on thermal efficiency has been shown in Fig. 16.32. The thermal efficiency initially increases, becomes maximum, and then starts decreasing. Figure 16.33 shows the effect of temperature ratio on thermal efficiency. It may be observed that the thermal efficiency increases as temperature ratio increases.

It is possible to use two turbines in parallel by using a single compressor and a single combustion chamber. The arrangement for such a system is shown in Fig. 16.34. For 1 kg of air through the compressor, a certain fraction x is required to pass through turbine I in order to drive the compressor. Then $(1 - x)$ kg passes through turbine II to produce the power output. This is advantageous in that the speed of the compressor would not be affected by the load. A further advantage in this respect can be had by the use of two combustion chambers as shown in Fig. 16.35.

In such an arrangement, the inlet temperatures of the two turbines, as well

as their speeds, are separately controllable.

Figure 16.34 *Two turbine in parallel*

Figure 16.35 *Two turbines of parallel with two combustion chambers*

Figure 16.36 *Two turbines in series*

16.15 ❐ MULTI-SHAFT SYSTEM TURBINES IN SERIES

The arrangement shown in Fig. 16.36 has two shafts with two turbines in series. It gives advantage of independent shaft speeds.

16.16 ❐ GAS TURBINE FUELS

The various fuels used for gas turbines are as follows:

1. **Gaseous fuels:** Natural gas, blast furnace, and producer gas
2. **Liquid fuels:** Liquid fuels of petroleum origin such as distillate oils or residual oils
3. **Solid fuels:** Pulverised coal and crushed coal

16.17 ❐ BLADE MATERIALS

Materials used in the manufacture of the blades consist of mainly three different elements, namely iron, nickel, and cobalt with chromium which forms one of the major alloying elements, since it gives high resistance to oxidation. Other alloying elements that have been used include most of the metals of the periodic table.

Materials working under high stress and high temperature have a particular rate of creep. It means elongation under stress is not a fixed quantity as it is in case of normal temperature. Thus, elongation continues to increase with time and the blades gradually take up the original gap provided at their tips. Therefore, contact with casing results in

failure. The repeated heating and cooling of the material affect the physical properties of the material.

16.17.1 Selection

The turbine blades are the most conditioned members of a gas turbine. They must withstand the following:

1. Effects of high operating temperature
2. Centrifugal tensile stresses due to rotational speeds of the order of 8000 to as high as 30,000 r.p.m.
3. Bending stress due to equivalent impulse load of the gases acting at a certain distance from fixing of the cantilever blade.
4. Hot erosive and corrosive effects due to high temperature combustion products *e.g.*, CO_2 and CO with O_2.

Turbine blade material must, therefore, be selected from considerations of the working conditions.

16.17.2 Requirements of Blade Material

The material should possess the following characteristics:

1. Maximum strength at high temperature
2. High creep strength
3. High resistance of corrosion
4. Maximum erosion resistance
5. Structural stability when exposed to varying temperatures
6. Castability or forgability
7. Weldability, if welding is used in the manufacture
8. Machinability
9. Absence of embrittlement in the service

16.18 ❒ COOLING OF BLADES

The thermodynamic analysis of the gas cycle shows that the efficiency of the cycle depends on the range of maximum temperature of the gas. If maximum temperature is increased, efficiency of the cycle is also increased. Any increase in the maximum temperature of gas results in an increase in the blade temperature, thereby inducing more thermal stresses in the blade material. Thus, this limits the capacity of gas temperature when blades are cooled properly.

16.18.1 Advantages of Cooling

1. Specific thrust of jet engine increases by 32% for gas temperature increase from 800°C to 1100°C at the expense of 7% specific fuel consumption.
2. If the maximum temperature is increased to 1600°C in place of present practice of 900°C, the specific fuel consumption will be increased by 50% and there is an increase in specific power output by 200%.

16.18.2 Different Methods of Blade Cooling

1. **Internal air cooling:** It is done by supplying cold air through hollow blade. It employs an internal baffle or deflector to direct the flow over the hotter portions on the internal surface (Fig. 16.37).
2. **Film cooling:** It involves supplying of thin film of cold compressed air through a narrow slit on the blade's surface to form a cold boundary layer over the blades (Fig. 16.38).
3. **Water cooling:** Circulation of water is maintained through a section of the blade from the root towards the tip.
4. **Rim cooling:** The rim is cooled by circulating the cooling fluid and thus the blade temperature is reduced by conduction.

Figure 16.39 shows the distribution of temperature over the blade height.

Figure 16.37 *Internal cooling of blade*

Figure 16.38 *Air film cooling of blade*

Figure 16.39 *Distribution of temperature over blade height*

Example 16.18

A gas turbine unit receives air at 100 kPa and 300 K and compresses it adiabatically to 620 kPa. The fuel has heating value of 44180 kJ/kg and fuel/air ratio is 0.017 kg fuel per kg air. The isentropic efficiencies of the compressor and the turbine are 88% and 90%, respectively. Calculate the compressor work, the turbine work, and the thermal efficiency.

Solution

The *T-s* diagram is shown in Fig. 16.40.

Given: p_1 = 100 kPa, T_1 = 300 K, p_2 = 620 kPa, C.V. = 44180 kJ/kg,

F:A = 0.017 kg fuel/kg air, η_c = 0.88, η_t = 0.90

Refer to Fig. 16.40.

Isentropic process 1–2:

or T_2 = 300 × 1.6842 = 505.3 K

Compressor efficiency,

or $T_{2'}$ = 533.3 K

Combustion process 2′–3:

$$\dot{m}_f \times \text{C.V.} = \dot{m}_a \times c_{pa} (T_3 - T_{2'}) + \dot{m}_f \times c_{pg} (T_3 - T_{2'})$$

Let c_{pa} = 1.005 kJ/kgK and c_{pg} = 1.15 kJ/kgK

$$0.017 \times 44180 = (1 \times 1.005 + 0.017 \times 1.15)\,(T_3 - 533.3)$$
$$= 1.0246\,(T_3 - 533.3)$$

Figure 16.40 *T-s diagram for gas turbine*

or T_3 = 1266.4 K

Isentropic process 3–4:

Turbine efficiency,

or

or $T_{4'}$ = 803.3 K

Compressor work, $w_c = c_{pa} (T_{2'} - T_1)$ = 1.005 (533.3 – 300) = 234.47 kJ/kg

Turbine work,

= (1.005 + 0.017 × 1.15) (1266.4 – 803.3) = 477.47 kJ/kg

Net work, $w_{net} = w_t - w_c$ = 474.47 – 234.47 = 240 kJ/kg

Heat supplied, $q_s = \dot{m}_f \times$ C.V. = $0.017 \times 44180 = 751.06$ kJ/kg

Thermal efficiency,

Example 16.19

The pressure ratio of an open cycle constant pressure gas turbine plant is 6. The temperature range of the plant is 20°C and 850°C. Using the following data; $c_{pa} = 1$ kJ/kgK, $c_{pg} = 1.05$ kJ/kgK, $\gamma_a = \gamma_g = 1.4$, C.V. of fuel = 44,000 kJ/kg, $\eta_c = 0.85$, $\eta_t = 0.90$, $\eta_{\text{comb}} = 0.95$, find (a) the thermal efficiency of the plant, (b) the net power developed if circulation of air is 5 kg/s, (c) the air-fuel ratio, and (d) the specific

fuel consumption.

Solution

Given: $r_p = 6$, $T_1 = 273 + 20 = 293$ K, $T_3 = 273 + 850 = 1223$ K, $c_{pa} = 1$ kJ/kgK

$c_{pg} = 1.05$ kJ/kgK, $\gamma_a = \gamma_g = 1.4$, $CV = 44,000$ kJ/kg,

$\eta_c = 0.85$, $\eta_t = 0.90$, $\eta_{\text{comb}} = 0.95$, $\dot{m}_a = 5$ kg/s

Refer to Fig. 16.40

Isentropic process 1–2:

or

or $T_{2'} = 523.43$ K

Combustion process 2′–3

$$\dot{m}_f \times \text{C.V.} \times \eta_{\text{comb}} = (\dot{m}_a \times c_{pa} + \dot{m}_f \times c_{pg})\,(T_3 - T_{2'})$$

Isentropic process 3–4:

Compressor work, $w_c = c_{pa}\,(T_{2'} - T_1) = 1 \times (523.43 - 293) = 230.43$ kJ/kg

Turbine work, w_t =

Net work per kg of air, $w_{\text{net}} = w_t - w_c = 411.19 - 230.43 = 180.76$ kJ/kg

Heat supplied,

1. Thermal efficiency of plant,
2. Net power developed,

$$P = \dot{m}_a \times w_{net} = 5 \times 180.76 = 903.8 \text{ kW}$$

3. Specific fuel consumption,

Example 16.20

Determine the efficiency of a gas turbine plant fitted with a heat exchanger of 75% effectiveness. The pressure ratio is 4:1 and the compression is carried out in two stages of equal pressure ratio with intercooling back to initial temperature of 290 K. The maximum temperature is 925 K. The isentropic efficiency of the turbine is 88% and the isentropic efficiency of each compressor is 85%. For air $\gamma = 1.4$, $c_p = 1.005$ kJ/kgK.

Solution

The T-s diagram is shown in Fig. 16.41.

Given: $T_6 = 925$ K, $\eta_t = 0.88$, $\eta_c = 0.85$, $\gamma = 1.4$, $c_p = 1.005$ kJ/kgK.

Refer to Fig. 16.41.

L.P. compressor:

or $T_{2'} = 364.7$ K

$\therefore T_{4'} = 364.7$ K for the H.P. compressor.

H.P. turbine:

Figure 16.41 *T-s diagram for gas turbine with intercooling*

or $T_{7'} = 560.3$ K

Heat exchanger effectiveness,

or

or $T_{5'} = 511.4$ K

Heat supplied, $q_s = c_p (T_6 - T_{5'}) = 1.005 (925 - 511.4) = 415.67$ kJ/kg

Compressor work, $w_c = 2c_p (T_{2'} - T_1) = 2 \times 1.005 (364.7 - 290) = 150.147$ kJ/kg

Turbine work, $w_t = c_p (T_6 - T_{7'}) = 1.005 (925 - 500.3) = 366.52$ kJ/kg

Net work, $w_{\text{net}} = w_t - w_c = 366.52 - 150.147 = 216.376$ kJ/kg

Efficiency of the gas turbine,

Example 16.21

Air enters the compressor of a gas turbine plant operating on Brayton cycle at 1 bar and 27°C. The pressure ratio in the cycle is 6. Assuming the turbine work 2.5 times the compressor work, calculate the maximum temperature in the cycle and cycle efficiency. Take $\gamma = 1.4$

Solution

Given: p_1 = 1 bar, T_1 = 273 + 27 = 300 K, r_p = 6, w_t = 2.5 w_c

Refer to Fig. 16.4 (b)

Process 1–2:

$$T_2 = T_1(r_p)^{(\gamma - 1)/\gamma} = 300(6)^{0.4/1.4} = 500.55 \text{ K}$$
$$w_c = c_p (T_2 - T_1) = 1.005(500.55 - 300) = 201.55 \text{ kJ/kg}$$

Process 3–4:

Now 0.40266 T_3 = 2.5 × 201.55

or T_3 = 1251.36 K

Maximum temperature is 1251.36 K.

Turbine work w_t = 2.5 × 201.55 = 503.87 kJ/kg

Net work, $w_{net} = w_t - w_c = 503.87 - 201.55 = 302.32$ kJ/kg

Heat supplied, $q_s = c_p\,(T_3 - T_2) = 1.005(1251.36 - 500.55)$

$$= 754.564 \text{ kJ/kg}$$

Cycle efficiency,

Example 16.22

A gas turbine cycle takes in air at 25°C and atmospheric pressure. The compression pressure ratio is 4. The isentropic efficiency of the compressor is 75%. The inlet temperature to turbine is limited to 750°C. What turbine efficiency would give overall cycle efficiency

zero percent?

Solution

Given: $T_1 = 273 + 25 = 298$ K, $p_1 =$ 1.013 bar,

$r_p = 4$, $\eta_c = 0.75$, $T_3 = 273 + 750 =$ 1023 K

Refer to Fig. 16.40,

Process 1–2:

$$T_2 = T_1\,(r_p)^{(\gamma-1)/\gamma} = 298(4)^{0.4/1.4} = 442.8 \text{ K}$$

or $T_{2'} = 491$K

Process 3–4:

$$T_{4'} = 1023 - 334.6\,\eta_t$$

Compressor work, $w_c = c_p\,(T_{2'} - T_1)$

$= 1.005(491 - 298) = 193.96$ kJ/kg

Turbine work, $w_t = c_p (T_3 - T_{4'}) = 1.005(1023 - 1023 + 334.6\ \eta_t)$

$= 336.27\ \eta_t$

Net work, $w_{net} = w_t - w_c = 336.27\ \eta_t - 193.96$

Heat supplied, $q_s = c_p (T_3 - T_{2'}) = 1.005(1023 - 491) = 534.66$ kJ/kg

Cycle efficiency,

or $336.27\ \eta_t - 193.96 = 0$

or $\eta_t = 0.5768$ or 57.68%

Example 16.23

In a Brayton cycle gas turbine power plant, the minimum and maximum temperatures of the cycle are 300 K and 1200 K. The compression is carried out in two stages of equal pressure ratio with intercooling of the working fluid to the minimum temperature of the cycle after the first stage of compression. The entire expansion is carried out in one stage only. The isentropic efficiency of both compressors is 0.85 and that of the turbine is 0.9. Determine the overall pressure ratio that would give the maximum net work per kg working fluid. Derive the expression you use. $\gamma = 1.4$.

[IES, 1995]

Solution

Given: $T_1 = T_3 = 300$ K, $T_5 = 1200$ K, , $\eta_{c1} = \eta_{c2} = 0.85$, $\eta_t = 0.90$, $\gamma = 1.4$

The $T-s$ diagram is shown in Fig. 16.42.

Figure 16.42 *T-s diagram for gas turbine*

Compressor 1:

Compressor 2:

Compressor work, $w_c = w_{c1} + w_{c2} = c_p\,[(T_2 - T_1) + (T_{4'} - T_1)]$

Turbine:

Turbine work,

Net work done, $w_{\text{net}} = w_t - w_c$

For maximum net work,

Example 16.24

In an open-cycle gas turbine plant, air enters at 15°C and 1 bar, and is compressed in a compressor to a pressure ratio of 15. The air from the exit of compressor is first heated

in a heat exchanger which is 75% efficient by turbine exhaust gas and then in a combustor to a temperature of 1600 K. The same gas expands in a two-stage turbine such that the expansion work is maximum. The exhaust gas from HP turbine is reheated to 1500 K and then expands to LP turbine. The isentropic efficiencies of compressor and turbine may be taken as 86% and 88%, respectively. The mechanical efficiencies of compressor and turbine are 97% each. The alternator efficiency is 98%. The output of turbo-alternator is 250 MW. Sketch the system and show the process on T-s diagram, Determine (*a*) the cycle thermal

efficiency, (*b*) the work ratio, (*c*) the specific power output, and (*d*) the mass flow rate of air.

[IES, 2007]

Solution

The schematic is shown in Fig. 16.43(a) and *T*-*s* diagram in Fig. 16.43(b).

1. $\eta_c = 0.86$, $\eta_t = 0.88$, $\eta_{alt} = 0.98$, $(\eta_{mech})_c = (\eta_{mech})_t = 0.97$

Figure 16.43 *Open cycle gas turbine: (a) Schematic, T-s diagram*

Compressor work,

For maximum expansion work,

$$p_5 = p_i = 3.873 \text{ bar}$$

$T_5 = = 1600 = 1086.7$ K

$\eta_{t1} =$

or $T_{5'} = T_4 - \eta_{t1} (T_4 - T_5)$

$$= 1600 - 0.88 (1600 - 1086.7) = 1158.5 \text{ K}$$

$T_7 = T_6$

$$p_7 = p_1 = 1 \text{ bar}, p_6 = p_i = 2.873 \text{ bar}$$

$T_7 = 1500 = 1018.8$ K

$\eta_{t2} =$

or $T_{7'} = 1500 - 0.88\ (1500 - 1018.8) = 1076.5$ K

Turbine work,

$$\begin{aligned} w_t &= w_{t1} + w_{t2} \\ &= 1.005\ [(1600 - 1158.5) + (1500 - 1076.5)] \times 0.97 \\ &= 843.245 \text{ kg} \end{aligned}$$

Net work, $w_{\text{net}} = w_t - w_c$

$$= 843.245 - 405.2 = 438.045 \text{ kJ/kg}$$

Heat exchanger effectiveness =

or 0.75 =

or $T_{3'} = 977.15$ K

Heat supplied = 1.005 [(1600 – 977.15) + (1500 – 1158.5)] = 969.17 kJ/kg

Thermal efficiency,

$\eta_{\text{th}} = = 0.452$ or 45.2%

2. Work ratio = = 0.5195.

3. Specific power output

$1 \times w_{net} = 1 \times 438.045 = 438.045$ kW

4. Mass flow rate of air =

= = 559.3 kg/s

Example 16.25

A gas turbine utilises a two-stage centrifugal compressor. The pressure ratios for the first and second stages are 2.5 and 2.1, respectively. The flow of air is 10 kg/s, this air being drawn at 1.013 bar and 20°C. If the temperature drop in the intercooler is 60°C and the isentropic efficiency is 90% for each stage, calculate:(*a*) the actual temperature at the end of each stage and (*b*) the total compressor power.

Assume $\gamma = 1.4$ and $c_p = 1.005$ kJ/kgK for air.

[IES, 2012]

Solution

Given: $r_{p1} = 2.5$, $r_{p2} = 2.1$, $\dot{m}_a = 10$ kg/s, $p_1 = 1.013$ bar, $T_1 = 20 + 273 = 293$ K, $\Delta T_2 = 60°C$, $\eta_{i1} = \eta_{i2} = 0.9$, $\gamma = 1.4$, $c_p = 1.005$ kJ/kgK.

The T–s diagram is shown in Fig. 16.44.

1. = 1.299

or $T_2 = 380.68$ K

$\eta_{i1} =$

or $0.9 =$

or $T_{2'} = 390.42$ K or 117.42°C

Temperature at the end of first stage is

117.42°C.

Temperature at the inlet of second stage

$$= 117.42 - 60 = 57.42°C$$
$$T_3 = 57.42 + 273 = 330.42 \text{ K}$$

Figure 16.44 *T-s diagram*

or $T_4 = 408.44$ K

$\eta_{i2} =$

0.9 =

Temperature at the end of second stage,

$$T_{4'} = 417.11 \text{ K or } 144.1°C$$

2. $P_{c1} = \dot{m}_a c_p (T_{2'} - T_1) = 10 \times 1.005 (390.42 - 293) = 979.07$ kJ/s

$P_{c2} = \dot{m}_a c_p (T_{4'} - T_3) = 10 \times 1.005 (417.11 - 330.42) = 877.23$ kJ/s

Total compressor power,

$P_c = P_{c1} + P_{c2} = 979.07 + 871.23 = 1850.3$ kW

Example 16.26

An open cycle gas turbine takes in

air at 300 K and 1 bar and develops a pressure ratio of 20. The turbine inlet temperature is 1650 K. The polytropic efficiency of the compressor and the turbine each is 90%. The pressure loss in the combustor is 3% and the alternator efficiency is 97%. Take c_{pa}= 1.005 kJ/kgK and c_{pg} = 1.128 kJ/kgK for air and gas, respectively. The calorific value of fuel is 42 MJ/kg. Determine the overall efficiency, the specific power output, the fuel to air ratio and the specific fuel consumption.

[IAS, 2010]

Solution

Given: T_1 = 300 K, p_1 = 1 bar, r_p =

20, T_3 = 1650 K, $h_c = h_t$ = 0.90, Δp_2 = 3%, h_{alt} = 0.97, c_{pa} = 1.005 kJ/kgK, c_{pg} = 1.128 kJ/kgK, C.V. = 42 MJ/kg

The T–s diagram is shown in Fig. 16.45.

Compressor process 1–2′:

or $T_2 = 300 \times 2.3535 = 706$ K

$\eta_c =$

or $0.9 =$

or $T_{2'} = 747.8$ K

$p_2 = 20 \times 1 = 20$ bar
$p_3 = 0.97 \times 20 = 19.4$ bar

Combustion process 2′–3:

$$\dot{m}_f \times CV = \dot{m}_f \times c_{pa} (T_3 - T_{2'}) + \dot{m}_f \times c_{pg} (T_3 - T_{2'})$$
$$= (\dot{m}_a \times c_{pa} + \dot{m}_f \times c_{pg}) (T_3 - T_{2'})$$

Figure 16.45 *T-s diagram*

or $\dot{m}_f \times 42 \times 10^3 = (\dot{m}_a \times 1.005 + \dot{m}_f \times 1.128) (1650 - 747.8)$

$$= 906.7 \, \dot{m}_a + 1017.7 \, \dot{m}_f$$

or $40982.3 \, \dot{m}_f = 906.7 \, \dot{m}_a$

Turbine process 3–4′:

$\gamma_g =$, $R = c_{pg} - c_{vg}$, $c_{vg} = c_{pg} - R = 1.128 - 0.287 = 0.841$ kJ/kgK

$\gamma_g = = 1.34$

$T_4 = = 777.5$ K

$\eta_t =$

or 0.9 =

or $T_{4'} = 864.75$ K

Specific power output:

$$\begin{aligned} w_{\text{net}} = w_t - w_c &= c_{pg} (T_3 - T_{4'}) - c_{pa} (T_{4'} - T_1) \\ &= 1.128 (1650 - 864.75) - 1.005 (747.8 - 300) \\ &= 435.72 \text{ kJ/kg of air} \end{aligned}$$

Overall efficiency, η_{overall} = =0.4281 or 42.81%

Specific fuel consumption =

= = 0.1828 kg/kWh

Summary for Quick Revision

1. Gas turbines may be classified as: (i) constant pressure combustion (a) open cycle type (b) closed cycle type and (ii) constant volume combustion

2. Brayton cycle:
    1. Compressor work, $w_c = c_p (T_2 - T_1)$
    2. Heat supplied, $q_s = c_p (T_3 - T_2)$
    3. Turbine work, $w_t = c_p (T_3 - T_4)$
    4. Net work, $w_{net} = w_t - w_c$
    5. Thermal efficiency, $\eta_{th} =$
    6. Specific output =
    7. For maximum specific output, $T_2 = T_4$
    8. Maximum work output, $(w_{net})_{max} =$
    9. Work ratio, R_w
    10. Optimum pressure ratio for maximum specific work output,
    11. $r_p =$

    and

3. Cycle operation with machine efficiencies:
    1. Optimum pressure ratio for maximum specific work output

        Isentropic efficiency of compressor,

        Isentropic efficiency of turbine,

        For specific work output to be maximum for given temperature limits,

        Pressure ratio,

    2. Optimum pressure ratio for maximum cycle efficiency,
4. Open cycle constant pressure gas turbine:

$$W_c = \dot{m}_a c_{pa} (T_{2'} - T_1)$$

$$W_t = (\dot{m}_a + \dot{m}_f) c_{pg} (T_3 - T_{4'})$$

$$W_{net} = w_t - w_c$$

$$Q = \dot{m}_f \times \text{C.V.}$$

$\eta_{th} =$

5. The thermal efficiency of open cycle gas turbine can be improved by: Regeneration, intercooling, reheating, and combination of above.
    1. Regeneration.
        1. Without machine efficiencies: $T_5 = T_4$

$$\eta_{\text{th}} = 1 - , c = , \alpha =$$

        2. With machine efficiencies.

$$\eta_{\text{th}} = 1 -$$

        3. Effectiveness of regenerator, $\in =$
    2. Intercooling:

For minimum compressor work.

$$w_c = c_{pa} (T_{2'} - T_1) + c_{pa}(T_{4'} - T_3)$$

Intermediate pressure for minimum compressor work,

$$p_i =$$

For $T_3 = T_1$ and $\eta_{c1} = \eta_{c2}$, $p_i =$

$$(w_c)_{\min} =$$

    3. Reheating: For $\eta_{t1} = \eta_{t2} = \eta_t$ and $T_5 = T_3$, $p_i =$ for maximum w_t.

$$(w_c)_{\max} = 2c_{pg}\ \text{h}_t\ T_3$$

Multiple-choice Questions

1. Consider the following statements:
    1. Intercooling is effective only at lower pressure ratios

and high turbine inlet temperatures.
2. There is very little gain in thermal efficiency when intercooling is used without the benefit of regeneration.
3. With higher value of γ and c_p of the working fluid, the net power output the Brayton cycle will increase.

Of these statements,

1. I, II, and III are correct
2. I and II are correct
3. I and III are correct
4. II and III are correct

2. In a gas turbine cycle with two stages of reheating, working between maximum pressure $p1$ and minimum pressure p_4 the optimum reheat reassure would be
    1. $(p_1p_4)^{1/2}$ and $(p_1p_2)^{2/3}$
    2. and
    3. and
    4. and
3. Intercooling in gas turbines
    1. decreases net output but increases thermal efficiency
    2. increases net output but decreases thermal efficiency
    3. decreases both net output and thermal efficiency
    4. increases both net output and thermal efficiency
4. Figure 16.46 shows four plots A, B, C, and D of thermal efficiency against pressure ratio.

Figure 16.46

The curve which represents that of a gas turbine plant using Brayton cycle (without regeneration) is the one labelled

1. A
2. B
3. C
4. D

5. The optimum intermediate pressure p_i for a gas turbine plant operating between pressure limits p_1 and p_2 with perfect intercooling between the two stages of compression (with identical isentropic efficiency) is given by

1. $p_1 = p_2 - p_1$
2.
3.
4.

6. In a gas turbine cycle, the turbine output is 600 kJ/kg, the compressor work is 400 kJ/kg and the heat supplied is 1000 kJ/kg. The thermal efficiency of this cycle is:
   1. 80%
   2. 60%
   3. 40%
   4. 20%
7. In a single-stage open-cycle gas turbine, the mass flow through the turbine is higher than the mass flows through compressor, because
   1. the specific volume of air increases by use of intercooler
   2. the temperature of air increases in the reheater
   3. the combustion of fuel takes place in the combustion chamber
   4. the specific heats at constant pressure for incoming air and exhaust gases are different
8. A gas turbine develops 120 kJ of work while the compressor absorbed 60 kJ of work and the heat supplied is 200 kJ. If a regenerator which would recover 40% of the heat in the exhaust were used, then the increase in the overall thermal efficiency would be:
   1. 10.2%
   2. 8.6%
   3. 6.9%
   4. 5.7%
9. A gas turbine works on which one of the following cycles?
   1. Brayton
   2. Rankine
   3. Stirling
   4. Otto
10. Reheating in a gas turbine
    1. increases the compressor work
    2. decreases the compressor work
    3. increase the turbine work
    4. decreases the turbine work
11. Consider the following statements relating to a closed gas turbine cycle:
    1. The cycle can employ monatomic gas like helium

    instead of air to increase the cycle efficiency if other conditions are the same.
2. The efficiency of heat exchanger increases with the use of helium.
3. The turbine blades suffer higher corrosion damages.
4. Higher output can be obtained for the same size.

Which of these statements are correct?

1. I, II, and III
2. I, II, and IV
3. II, III, and IV
4. I, III, and IV

12. The efficiency of a simple gas turbine can be improved by using a regenerator, because the
    1. work of compression is reduced
    2. heat required to be supplied is reduced
    3. work output of the turbine is increased
    4. heat rejected is increased
13. In a regenerative feed heating cycle, the economic number of the stages of regeneration
    1. increases as the initial pressure and temperature increase
    2. decreases as the initial pressure and temperature increase
    3. is independent of the initial pressure and temperature
    4. depends only on the condenser pressure
14. Match List I (Blades) with List II (Features) and select the correct answer using the codes given below the Lists:

Codes:

A B C D

1. 2 1 5 4
2. 3 4 5 1
3. 2 4 3 5
4. 3 2 1 5

15. Consider the following statements about modification in a gas

turbine power plant working on a simple Brayton cycle:

1. Incorporation of the regeneration process increases specific work output as well as thermal efficiency.
2. Incorporation of regeneration process increases thermal efficiency but specific work output remains unchanged.
3. Incorporation of intercooling process in a multi-stage compression increases specific work output but the heat input also increases.
4. Incorporation of intercooling process in a multi-stage compression system increases specific work output, the heat addition remains unchanged.

Which of the above statements are correct?

1. 1 and 3
2. 1 and 4
3. 2 and 3
4. 2 and 4

16. A power plant, which used a gas turbine followed by steam turbine for power generation, is called
    1. Topping cycle
    2. Bottoming cycle
    3. Brayton cycle
    4. Combined cycle
17. Consider the following statements:
    1. The speed of rotation of the moving elements of gas turbines is much higher than those of steam turbines.
    2. Gas turbine plants are heavier and larger in size than steam turbine plants.
    3. Gas turbines require cooling water for its operations.
    4. Almost any kind of fuel can be used with gas turbines.

Which of the statements given above are correct?

1. I and II
2. I and III
3. I and IV
4. III and IV

18. The thermal efficiency of a gas turbine cycle with regeneration in terms of T_3 (maximum temperature), T_1 (minimum temperature), r_p (pressure ratio) and is given by

1.
2.
3.
4.

19. Consider the following features for a gas turbine plant:
    1. Intercooling
    2. Regeneration
    3. Reheat

    Which of the above features in a simple gas turbine cycle increase the work ratio?

    1. I, II, and III
    2. Only I and II
    3. Only II and III
    4. Only I and IV
20. Consider the following statements:

    For a large aviation gas turbine, an axial flow compressor is usually preferred over centrifugal compressor because

    1. the maximum efficiency is higher
    2. the frontal area is lower
    3. the pressure rise per stage is more
    4. the cost is lower

    Which of the statements given above are correct?

    1. I and IV
    2. Only I and II
    3. I, II, and III
    4. II, III, and IV
21. In a reaction turbine, the heat drop in fixed blade is 8 kJ/kg and the total heat drop per stage is 20 kJ/kg. The degree of reaction is
    1. 40%
    2. 66.7%
    3. 60%
    4. 25%
22. Consider the following statements:

Which of the following increase the work ratio in a simple gas turbine plant?

1. Heat exchanger
2. Inter cooling
3. Reheating

Select the correct answer using the code given below:

1. 1 and 2
2. 2 and 3
3. 1 and 3
4. 1, 2, and 3

23. Optimum pressure ratio for maximum specific output for ideal gas turbine plant operating at initial temperature of 300 K and maximum temperature of 1000 K is closer to
    1. 4
    2. 8
    3. 12
    4. 16
24. The cycle shown in Fig. 16.47 represents a gas turbine cycle with intercooling and reheating.

Figure 16.47

Match List-X (Units) with List-Y (Processes) and select the correct answer using the codes given below the Lists:

Codes:

A B C D

1. I II IV III
2. II IV V III

3. III IV V II
4. II V IV I

25. Which one of the following is the correct statement? In a two-stage gas turbine plant with intercooling and reheating,
    1. both work ratio and thermal efficiency improve
    2. work ratio improves but thermal efficiency decreases
    3. thermal efficiency improves but work ratio decreases
    4. both work ratio and thermal efficiency decrease

Explanatory Notes

1. 6. (a) Thermal efficiency = = 0.2 or 20%
2. 8. (d) Thermal efficiency = 0.3 or 30%

Heat in exhaust gas = 200 – 120 = 80 kJ

Heat recovered in regenerator = 80 × 0.4 = 32 kJ

Heat supplied = 200 – 32 = 168 kJ

New thermal efficiency 0.357 or 35.7%

Increase in efficiency = 35.7 – 30 = 5.7%

1. 21. (c) Degree of reaction
2. 23. (b) Optimum pressure ratio for maximum specific output of a gas turbine,

Review Questions

1. List the fields of application of gas turbines.
2. What are the limitations of a gas turbine?
3. List four advantages of gas turbine over an IC engine.
4. List four disadvantages of gas turbine over an IC engine.
5. Enumerate five advantages of gas turbine over steam turbine.
6. How do you classify gas turbines?

7. Make a comparison of open and closed cycle gas turbines.
8. Explain the position of the gas turbine in the power industry.
9. Write the formula for the thermal efficiency of the Brayton cycle.
10. Define specific output and work ratio.
11. Write the expressions for maximum work output and work ratio of the Brayton cycle.
12. Draw the schematic arrangement of open cycle constant pressure combustion gas turbine.
13. What do you understand by regeneration?
14. Define effectiveness of a regenerator.
15. What is the intermediate pressure for intercooling to have minimum compressor work?

Exercises

16.1 A Brayton cycle operates with air between 1 bar, 300 K and 5 bar, 1000 K. The air is compressed in two stages with perfect intercooling. Similarly, in the turbine, expansion occurs in two stages with perfect reheating. Calculate the optimum intermediate pressure, net work output, and the fraction of turbine output that has to be put back to the compressor.

[**Ans.** 2.236 bar, 257.3 kJ/kg, 0.378]

16.2 A gas turbine unit operates at a mass flow of 30 kg/s. Air enters the compressor at a pressure of 1 bar, 288 K and is discharged from the compressor at a pressure of 10.5 bar. Combustion occurs at constant pressure and results in a temperature rise of 420 K. If the flow leaves of the turbine at a pressure of 1.2 bar, determine the net power output from the unit and also the thermal efficiency.

[**Ans.** 5338.74 kW, 42.55%]

16.3 In a regenerative gas turbine cycle, air enters the compressor at a temperature and pressure of 30°C and 1.5 bar and discharges at 220°C and 5.2 bar. After passing through the regenerator, the air temperature is

395°C. The temperature of air entering and leaving the gas turbine are 900°C and 510°C, respectively. Assuming no pressure drop through the regenerator, determine (a) the output per kg of air, (b) the efficiency of cycle, and (c) the work required to drive the compressor. Take $c_p = 1.005$ kJ/kgK.

[**Ans.** 201 kJ/kg, 39.6%, 190.95 kJ/kg]

16.4 A Brayton cycle works between 1 atm, 27°C and 5 atm, 977°C. There are two stages of compression with perfect intercooling and two stages of expansion. The work output of first expansion stage being used to drive the two compressors, where the interstage pressure is optimised for the compressor. The air from the first stage

turbine is again heated to 977°C and expanded. Calculate the power output of free power turbine and cycle efficiency without and with a perfect heat exchanger and compare them. Calculate the percentage improvement in the efficiency because of the addition if heat exchangers.

[**Ans.** 350.8 kJ/kg, 33.96%, 69.2%, 50.9%]

16.5 The following data apply to a gas turbine set employing a separate power turbine, regenerator, and intercooler between two-stage compression:

Isentropic efficiency of compression in each stage = 80%

Isentropic efficiency of compressor turbine = 88%

Isentropic efficiency of power turbine =88%

Turbine to compressor transmission efficiency = 98%

Pressure ratio in each stage of compression = 3:1

Temperature after intercooler = 297 K

Air mass flow rate = 15 kg/s

Regenerator effectiveness =80%

Regenerator gas-side pressure loss = 0.1 bar

Maximum turbine temperature = 1000 K

Ambient temperature = 327 K

Ambient pressure = 1 bar

Calorific value of fuel = 43.1 MJ/kg

Calculate (a) the net power output, (b) specific fuel consumption, and (c) overall thermal efficiency. Assume that the pressure losses in the air-side of the regenerator and combustion chamber are accounted for in the compressor efficiency. Take c_{pa} = 1.005 kJ/kgK, c_{pa} = 1.147 kJ/kgK, γ_{α} = 1.4 and γ_g = 1.33.

[**Ans.** 1811.67 kW, 0.32 kg/kWh, 26.1%]

16.6 In a gas turbine plant, air is compressed from 1 bar, 15°C through a pressure ratio of 4:1. It is then heated to 650°C in a combustion chamber and expanded back to atmospheric pressure of 1 bar in a turbine. Calculate the cycle

efficiency and the work ratio if a perfect heat exchanger is used. The isentropic efficiencies of turbine and compressor are 85% and 80%, respectively. Take c_{pa} = 1.00 kJ/kgK.

[**Ans.** 31.6%, 0.316]

16.7 In a gas turbine plant, the pressure ratio through which air at 15°C is compressed is 6. The same air is then heated to a maximum permissible temperature of 750°C, first in a heat exchanger having effectiveness of 75%, and then in the combustion chamber. The same air at 750°C is expanded in two stages such that the expansion work is maximum. The air is reheated to 750°C after the first stage. Determine (a) the cycle thermal efficiency, (b) the

work ratio, and (c) the net shaft work per kg of air. The machine efficiencies may be assumed to be 80% and 85% for the compressor and turbine, respectively. Take c_{pa} = 1.00 kJ/kgK.

[**Ans.** 33.6%, 0.398, 160 kJ/kg]

16.8 In a gas turbine plant, the air at 283 K and 1 bar is compressed to 4 bar with compression efficiency of 80%. The air is heated in the regenerator and in the combustion chamber till its temperature is raised to 973 K, and during the process, the pressure falls by 0.1 bar. The air is then expanded in the turbine and passes to the regenerator which has 75% effectiveness and causes a pressure drop of 0.14 bar. If the isentropic efficiency of the turbine is 85%,

determine the thermal efficiency of the plant.

[**Ans.** 23.15%]

16.9 A gas turbine works on Brayton cycle between 27°C and 827°C. Determine the maximum net work per kg and the cycle efficiency. Take c_{pa} = 1.005 kJ/kgK.

[**Ans.** 252.34 kJ/kg, 47.77%]

16.10 Atmospheric air enters the compressor of an open cycle constant pressure gas turbine at a pressure of 1 bar, 293 K. The pressure of air after compression is 4 bar. The isentropic efficiencies of compressor and turbine are 80% and 85%, respectively. The air-fuel ratio used is 90:1. If the mass flow rate of air is 3 kg/s, then calculate (a) the

power developed and (b) the thermal efficiency of the cycle. Assume $c_{pa} = c_{pg} = 1.0$ kJ/kgK and $\gamma_a = \gamma_g = 1.4$. Calorific value of fuel = 41800 kJ/kg.

[**Ans.** 252.84 kW, 18.14%]

16.11 In a constant pressure open cycle gas turbine, air enters at 1 bar, 20°C and leaves the compressor at 5 bar. Temperature of gases entering the turbine is 680°C, pressure loss in the combustion chamber is 0.1 bar, $\eta_c = 0.85$, $\eta_t = 0.80$, $\eta_{\text{combustion}} = 0.85$, $g_a = 1.4$ and $c_{pa} = c_{pg} = 1.024$ kJ/kg.K. Calculate (a) the quantity of air circulated if power developed is 1065 kW, (b) the heat supplied per kg of air circulated, and (c) the thermal efficiency of the cycle. Neglect the mass of fuel as

compared to that of air.

[**Ans.** 13.43 kg, 552.9 kJ/kg, 14.34%]

16.12 An open cycle constant pressure gas turbine draws air at 1.01 bar, 15°C and pressure ratio of 7:1. The compressor is driven by the H.P. turbine and L.P. turbine drives a separate power shaft. The isentropic efficiencies of compressor, H.P. and L.P. turbines are 0.82, 0.85, and 0.85, respectively. If the maximum cycle temperature is 610°C, calculate (a) the pressure and temperature of gases entering the power turbine, (b) the net power developed (c) the work ratio, and (d) the thermal efficiency of the plant. Assume c_{pa} = 1.005 kJ/kgK, γ_a = 1.4; c_{pg} = 1.15 kJ/kgK, γ_g = 1.333. Neglect mass of fuel.

[**Ans.** 1.636 bar, 654.4 K; 72.1 kW; 0.25; 18.8%]

16.13 A gas turbine employs a heat exchanger with a thermal ratio of 0:72. The turbine operates between 1.01 bar and 4.04 bar and ambient temperature is 20°C. The isentropic efficiencies of the compressor and turbine are 80% and 85%, respectively. The pressure drop on each side of the heat exchanger is 0.05 bar and in the combustion chamber 0.14 bar. Assume $\eta_{\text{combustion}} = 1.0$ and. C.V. of fuel = 41800 kJ/kg. Calculate the increase in thermal efficiency due to heat exchanger.

16.14 A gas turbine has a pressure ratio of 6:1 and a maximum cycle temperature of 600°C. The isentropic efficiencies of the compressor and

turbine are 82% and 85%, respectively. Calculate the power output when the air enters the compressor at 15°C at the rate of 15 kg/s. Take c_{pa} = 1.005 kJ/kgK, γ_a = 1.4; c_{pg} = 1.11 kJ/kgK, γ_g = 1.333.

[**Ans.** 920 KW]

16.15 A gas turbine plant operates between the temperatures of 1200 K and 320 K and corresponding pressures of 6 bar and 1.2 bar. The ratio of turbine and compressor efficiencies is 0.95. Determine the two efficiencies if the plant operates at optimum pressure ratio for maximum specific output. Assume γ = 1.4.

[**Ans.** η_t = 79.72%, η_c = 83.90%]

16.16 The maximum and minimum temperatures occurring in a closed cycle

gas turbine plant are 927°C and 37°C. If the pressure at the outlet and the inlet of the compressor are 5 bar and 1 bar, respectively. Determine the compressor work, the turbine work, the heat supplied to the cycle, and the net work done in the cycle. Also determine the thermal efficiency of the cycle and the optimum pressure ratio for maximum power output for the given temperatures. Assume γ for air = 1.4 and c_p = 1.005 kJ/kgK.

16.17 An open cycle constant pressure gas turbine plant operates with a pressure ratio of 6.5. The temperatures at the inlet to the compressor and turbine are 17°C and 820°C, respectively. Assume: specific heat at

constant pressure of air and gas to be 1.005 kJ/kgK and 1.079 kJ/kgK, respectively; ratio of specific heats for air and gas to be 1.4; calorific value of fuel 44350 kJ/kg; efficiency of compressor and turbine to be 0.86 and 0.92, respectively. Calculate: (i) the power of the plant for air circulation of 6 kg/s, (ii) the thermal efficiency of plant, (iii) the air-fuel ratio, and (iv) the specific fuel consumption. Take the mass of the fuel into account.

16.18 A gas turbine power plant works between pressures of 1 bar and 5 bar and temperatures of 285 K and 1100 K. The intercooler cools the air at 2.3 bar to 285 K before the air is sent to the second stage compressor. The compressor air

from the second-stage compressor passes through a regenerator having effectiveness of 0.72 and then through the combustion chamber. The heated air is then expanded in a high pressure turbine to 2.3 bar and is then reheated to 1100 K. The air is finally expanded in the low-pressure turbine to 1 bar. Assuming the compressor and turbine efficiencies to be 85%, determine: (i) the ratio of compression work to the turbine work, (ii) the power developed for an air flow of 3 kg/s, (iii) the thermal efficiency of the cycle, (iv) the heat rejected per second to the cooling water in the intercooler, and (v) the heat rejected per second to the atmosphere. Assume that all the components are mounted on the same shaft. Sketch the

flow diagram of the turbine and represent the process on T-s plane. Also, assume c_p = 1.005 kJ/kg.K and g = 1.4.

16.19 A gas turbine set takes in air at 27°C and 1 atm. The pressure ratio is 4 and the maximum temperature is 560°C. The compressor and turbine efficiencies are 83% and 85%, respectively. If the regenerator effectiveness is 0.75, determine the overall efficiency.

ANSWERS TO MULTIPLE-CHOICE QUESTIONS

1. d
2. a
3. b
4. b
5. c
6. d
7. c
8. d
9. a
10. c
11. b
12. b
13. a
14. a
15. d

16. d
17. c
18. d
19. d
20. b
21. c
22. b
23. b
24. b
25. b

Chapter 17

Jet Propulsion

17.1 ❐ PRINCIPLE OF JET PROPULSION

Jet propulsion is the thrust produced by passing a jet of fluid in the direction opposite to the motion of the object. The propulsive thrust in jet propulsion is obtained by the reaction of the engine produced by the exit of high velocity fluid stream from the nozzle. In case of a jet engine, the atmospheric air is drawn into the engine and compressed in a compressor driven by a gas turbine. The compressed air is used in the combustion chamber of the gas turbine and is partly expanded to extract enough

work to drive the air compressor. The balance expansion takes place in the nozzle and the high velocity exhaust from the nozzle provides the thrust, which is used to propel the vehicles.

17.2 ❐ JET PROPULSION SYSTEMS

Jet propulsion systems may be classified as follows:

1. **Air stream jet engines:** In these engines, the oxygen necessary for combustion is taken from the atmosphere. They are also called air-breathing engines. They may be classified into the following:
    1. Screw propeller
    2. Ramjet engine
    3. Pulse jet engine
    4. Turbojet engine
    5. Turboprop engine
2. **Self-contained rocket engines:** In a rocket engine, the full and the oxidiser are contained in the propelling body and thus can function in vacuum as well. They are also called non-air breathing engines. They may be classified into the following:
    1. Liquid propellant
    2. Solid propellant

17.2.1 Screw Propeller

The line diagram of a screw propeller is

shown in Fig. 17.1(a). The schematic arrangement and the corresponding T-s diagram are shown in Fig. 17.1(b) and Fig. 17.1(c), respectively. In this system, full expansion takes place in the turbine, and the total power developed by the turbine is used to drive the compressor and the propeller. The power supplied to the propeller is controlled by controlling the supply of fuel in the combustion chamber. The rate of increase in efficiency of the screw propeller is higher at lower speed but its efficiency falls rapidly at higher speed. It has high power for take-off and higher propulsion efficiency at speed below 600 km/h.

Figure 17.1 *Screw propeller: (a) Line diagram, (b) Schematic diagram, (c) T-s diagram*

17.2.2 Ramjet Engine

The line diagram of a ramjet engine is shown in Fig. 17.2(a) and the corresponding *T*-s diagram is shown in Fig. 17.2(b). It consists of a supersonic diffuser, subsonic diffuser, combustor, and nozzle. Both supersonic and subsonic diffusers convert the kinetic energy of the entering air into pressure rise. This energy transformation is called the ram effect and the pressure rise is called the *ram pressure*.

Air from the atmosphere enters the engine at a very high speed and its velocity gets reduced first in the supersonic diffuser, thereby its static pressure increases. The air then enters the subsonic diffuser wherein it is compressed further and its pressure and

temperature reaches above the ignition temperature. Later, the air flows into the combustion chamber, the fuel is injected and mixed with the unburnt air. The high pressure and high temperature gases are passed through the nozzle, converting the pressure energy into kinetic energy. The high velocity gases leaving the nozzle provide the required forward thrust to the ramjet.

The major advantages of ramjet engines are as follows: light in weight, no moving parts, and possibility of using a wide variety of fuels. However, its main drawback is that it cannot be started on its own from rest and requires some launching device. Due to its high thrust at high operational speed, it is widely

used in high-speed military aircrafts and missiles. Subsonic ramjets are used in target weapons, in conjunction with turbojets or rockets for getting the starting torque. It can be used for pilotless aircraft.

Figure 17.2 *Ramjet engine: (a) Line diagram, (b) T-s diagram*

17.2.3 Pulse Jet Engine

The line diagram of a pulse jet engine is shown in Fig. 17.3. It consists essentially of a diffuser, valve grid, combustion chamber, spark plug, and a tail pipe or discharge nozzle. It is an intermittent combustion engine and operates similar to a reciprocating engine. It develops thrust by a high velocity of jet of exhaust gases without the aid of a compressor or turbine.

The incoming air is compressed by the ram effect in the diffuser and the grid passages which are opened and closed by V-shaped non-return valves. The fuel is then injected into the combustion chamber by fuel injectors. The combustion is then initiated by a spark plug. As a result of combustion, the temperature and pressure of combustion products increase and the non-return valves get closed; consequently, the hot gases flow out. The static pressure in the chamber falls and the high pressure air in the diffuser forces the valves to open and fresh air is admitted for the new cycle.

Figure 17.3 *Pulse-jet engine*

Its main advantages are that it is simple

in construction, very inexpensive as compared to turbo-jet engine, and well adaptable to pilotless aircrafts. It is also capable of producing static thrust.

Its major shortcomings are: high intensity of noise, severe vibrations, high rate of fuel consumption, low efficiency, intermittent combustion, limited operational attitude and serious limitations of mechanical valve arrangement.

17.2.4 Turbo-jet Engine

The line diagram of the turbojet unit is shown in Fig. 17.4(a). The schematic arrangement and the corresponding *T*-s diagram are shown in Fig. 17.4(b) and Fig. 17.4(c), respectively. It consists of a diffuser, compressor, combustion

chamber, turbine, and exhaust nozzle.

Figure 17.4 *Turbo-jet engine: (a) Line diagram, (b) Schematic diagram, (c) T-s diagram*

The function of the diffuser is to convert the kinetic energy of the entering air into a static pressure rise, which is achieved by the ram effect. This air enters the axial flow compressor and combustion chamber. The gas turbine produces just sufficient power to drive the compressor by partial expansion in the turbine. The gases coming out from the turbine are expanded in the nozzle and produce a very high velocity jet which gives a forward motion to the aircraft by the jet reaction.

The advantages of a turbo-jet engine are its simpler construction, absence of

vibrations, uninterrupted and smooth operation, lesser weight to power ratio, higher rate of climb, and higher propulsion efficiency at higher speed. However, its disadvantages are less efficiency, shorter life, noisier, expansive, larger length of take-off required, and low thrust at take-off.

It is suitable for piloted operation of aircraft travelling above 800 km/h.

17.2.5 Turbo-Prop Engine

The line diagram of a turbo-prop engine is shown in Fig. 17.5(a) and the corresponding T-s diagram in Fig. 17.5(b). The engine consists of propeller, reduction gear, diffuser, compressor, combustion chamber, turbine, and exhaust nozzle. In this

system, the gases are partly expanded in the turbine and partly in the nozzle. The total power produced by the turbine is used to run the compressor and the propeller. The propeller provides most of the propulsive thrust and the thrust produced due to jet action is quite small.

The turbo-prop combines the merits of a turbo-jet and a propeller. It is suitable for piloted operation of aircraft.

Figure 17.5 *Turboprop engine: (a) Line diagram, (b) T-s diagram*

Figure 17.6 *Line diagram of a rocket jet*

17.2.6 Rocket Propulsion

In a rocket engine, the fuel and the oxidiser are stored in the propelling body. The rocket can function in a vacuum and is the only device capable

of space flight.

Rockets may be of the single-stage or multi-stage type, consisting of one or more rocket motors. They may be solid propellant or liquid propellant rockets.

Figure 17.6 shows the line diagram of a liquid propellant rocket engine using liquid oxygen and refined petrol. It consists of a fuel tank, oxidiser tank, gas turbine, pumps, combustion chamber, and nozzle. Pumps are used to supply propellants to the nozzle at high pressure to obtain higher thrust. These pumps are operated by a gas turbine where the high pressure and high temperature gases formed by the fuel and oxidiser are delivered to the gas turbine.

The properties for an ideal propellant are high calorific value, high density, high stability, ease of handling and storing, non-corrosive, and high boiling point at low pressure.

Rockets find important applications in the field of long range artillery, lethal weapons, signalling and fireworks, jet assisted take-off, satellites, space ships, and research. They are suitable for pilotless operation.

17.3 ❐ JET PROPULSION V'S ROCKET PROPULSION

Table 17.1 shows the comparison of jet propulsion and rocket propulsion.

Table 17.1 *Comparison of jet and rocket propulsion*

The basic cycle of a turbo-jet engine in *T*-s diagram is shown in Fig. 17.7. The various processes are as follows:

Process 1–2: Isentropic diffusion of atmospheric air from velocity c_1 to $c_2 = 0$ with diffuser efficiency of 100%. 1–2′ is the actual diffusion with diffuser efficiency η_d from p_1 to p_2.

Process 2′–3: Isentropic compression of air in the compressor from p_2 to p_4. Process 2′–3′ is the actual compression with compressor efficiency η_c.

Process 3–4: Ideal heat addition at pressure p_4. Process 3′-4 is the actual heat addition at pressure p_4.

Process 4–5: Isentropic expansion of gas in the turbine from p_4 to p_3. Process 4–5′ is the actual expansion in the turbine with turbine efficiency η_t.

Process 5′–6: Isentropic expansion of gas in the nozzle from p_3 to p_1. Process 5′–6′ is the actual expansion of gas in the nozzle with nozzle efficiency η_n.

Consider 1kg of working fluid flowing through the system.

Diffuser:

The energy equation between states 1 and 2 is as follows:

In an ideal diffuser, $c_2 = 0$, $q_{1-2} = 0$ and $w_{1-2} = 0$

where c_{pa} = specific heat of air at constant pressure

Figure 17.7 *T-s diagram for turbo-jet*

Compressor:

Energy equation between state 2′ and 3:

Compressed work,

Assuming $c_3 \simeq c_{2'}$, and $q_{2'-3} = 0$, we have

$$w_c = h_3 - h_{2'} = c_{pa} (T_3 - T_{2'})$$

Actual compressor work, $w_{ca} = h_{3'} - h_{2'}$

Combustion chamber:

Ideal heat supplied, $q = q_3{}' - {}_4 = h_4 - h_3{}'$

where c_{pg} = specific heat of gas at constant pressure

Turbine:

Energy equation between states 4 and 5:

If $q_{4\text{-}5} = 0$, then

Turbine work,

If $c_4 \simeq c_5$, then

now $w_{ta} = w_{ca}$

If $c_{pa} = c_{pg} = c_p$, then

$$T_{5'} = T_4 - (T_4 - T_5)\eta_t$$

Jet nozzle:

Energy equation between states 5′ and 6:

Ideal case:

Actual case:

If $c_{5'} \simeq 0$ as compared to $c_{6'}$, then

Thermal efficiency,

17.4.1 Thrust

Let c_a = forward velocity of aircraft through air, m/s

= velocity of approach, assuming surrounding air velocity to be zero.

c_j = velocity of jet of gases relative to the aircraft, m/s

, mass rate of flow of products leaving the nozzle per 1kg of air.

Absolute velocity of gases leaving the aircraft = $c_j - c_a$

Absolute velocity of air entering the aircraft = 0

$\therefore$ Change of momentum = $\dot{m}\,(c_j - c_a)$

17.4.2 Thrust Power

It is defined as the rate at which work must be developed by the engine if the aircraft is to be kept moving at a constant velocity c_a against friction

force or drag.

Thrust power, T.P. = forward thrust × speed of aircraft

17.4.3 Propulsive Power

The energy required to change the momentum of the mass flow of gases represents the propulsive power. It is expressed as the difference between the rate of kinetic energies of the entering air and exit gases.

17.4.4 Propulsive Efficiency

It is defined as the ratio of thrust power to propulsive power.

17.4.5 Thermal Efficiency

It is defined as the ratio of propulsive work to the energy released by the combustion of fuel.

17.4.6 Overall Efficiency

Overall efficiency,

for η_0 to be maximum,

$$c_j - 2c_a = 0$$

Therefore, for maximum overall efficiency, the aircraft velocity is one-half of the jet velocity.

17.4.7 Jet Efficiency

Jet efficiency is defined as follows:

17.4.8 Ram Air Efficiency

In Fig. 17.2(b), process 1-2′ is the ramming process during which the total energy or stagnation enthalpy remains unchanged, if the process is assumed adiabatic, while pressure of air increases. Process 1-2 is the ideal ramming action.

17.5 ❐ THRUST WORK, PROPULSIVE WORK, AND PROPULSIVE EFFICIENCY FOR ROCKET ENGINE

Neglecting the friction and other losses, we have

Rocket propulsive efficiency

Example 17.1

A turbo-jet unit consists of single stage compressor, single stage turbine, and nozzle. The pressure and temperature at the inlet of compressor are 0.8 bar and 280 K, respectively. The pressure at the outlet of compressor is 4.5 bar. The following data is available:

Isentropic efficiency of compressor = 0.82

Maximum cycle temperature = 560°C

Isentropic efficiency of turbine = 0.85

Isentropic efficiency of nozzle = 0.90

Combustion efficiency = 0.98

Mechanical efficiency = 0.95

c_{pa} = 1.005 kJ/kg.K, γ_a = 1.4, c_{pg} = 1.2 kJ/kg.K, γ_g = 1.35

C.V. of fuel used = 42,000 kJ/kg

Static back pressure on the nozzle = 0.65 bar

Speed of aircraft = 720 km/h

Mass rate of flow of air = 20 kg/s

Calculate (a) I.P. required to drive the compressor, (b) A.F. ratio, (c)

thrust developed, and (d) the propulsion efficiency.

Solution

The T-s diagram is shown in Fig. 17.8.

Figure 17.8 *T-s diagram for turbo-jet*

$$T_3 = 560 + 273 = 833 \text{ K}$$

Heat supplied by the fuel = Heat carried by gases from combustion chamber

$$m_f \times \text{C.V.} \times \eta_{\text{comb}} = c_{pg} (\dot{m}_a + \dot{m}_f) (T_3 - T_{2'})$$

No work developed by turbine = work required to run the compressor

I.P. required to run the compressor $= \dot{m}_a \times c_{pa} (T_{2'} - T_1)$

$$= 20 \times 1.005 (497.8 - 280) = 4377.78 \text{ kW}$$

Heat drop in nozzle, $\Delta h = c_{pg} (T_{4'} - T_{5'})$

$$= 1.2 (642.87 - 543.3)$$
$$= 119.484 \text{ kJ/kg of air}$$

Velocity of gases learning the nozzle relative to aircraft,

Velocity of aircraft,

Mass of gases passing through the nozzle,

Thrust produced $= \dot{m}_g (c_j - c_a) =$

20.197 (488.84 – 200) = 5833.7 N

Propulsive efficienc

Example 17.2

In a jet propulsion system, air enters the compressor at 1 bar, 10°C. The pressure leaving the compressor is 4 bar and the maximum temperature is 85°C. The air expands in the turbine to such a pressure that the turbine work is just equal to the compressor work. On leaving the turbine, the air expands in a nozzle to 0.8 bar. Calculate the velocity of air leaving the nozzle. Take c_p = 1.005kJ/kg.K and γ = 1.4 for both compression and expansion

processes.

Solution

The T-s diagram is shown in Fig. 17.9.

Compressor work, $w_c = c_p \, (T_2 - T_1)$

$$= 1.005\,(420.5 - 283) = 138.2 \text{ kJ/kg of air}$$
$$T_3 = 273 + 850 = 1123 \text{ K}$$

Figure 17.9 *T–s diagram for jet propulsion*

Now $w_t = w_c = c_p \, (T_3 - T_4)$

or $1.005(1123 - T_4) = 138.2$

or $T_4 = 985.46$ K

Assuming $c_4 \simeq 0$, $\Delta h = c_p \, (T_4 - T_5)$
$= 1.005\,(985.46 - 709) = 277.8$ kJ/

kg

Example 17.3

A turbojet engine inducts 40 kg/s of air and propels an aircraft with a uniform flight speed of 900 km/h. The isentropic enthalpy change for nozzle is 190 kJ/kg and its velocity coefficient is 0.96. The fuel-air ratio is 0.015. The combustion efficiency is 0.95 and the lower heating value of the fuel is 44,000 kJ/kg. Calculate the thermal efficiency of the engine, the fuel flow rate and TSFC, the propulsive power, the thrust power, the propulsive efficiency, and the overall

efficiency.

Solution

Jet velocity,

Aircraft velocity,

$$\dot{m}_f = 0.015\ m_a = 0.015 \times 40$$
$$= 0.6 \text{ kg/s or } 2160 \text{ kg/h}$$

Thrust, F = $(\dot{m}_a + \dot{m}_f)\ c_j - \dot{m}_a\ c_a$

= (40 + 0.6) × 585.6 – 40 × 250

= 13775 N or 13.775 kN

Thrust specific fuel consumption,
TSFC

Propulsion power, P.P.

$$= 5711 \times 10^3 \text{ W} = 5711 \text{ kW}$$

Thrust power, T.P. = $[(\dot{m}_a + \dot{m}_f)cj - \dot{m}_a c_a] \times c_a$

$$= (40.6 \times 585.6 - 40 \times 250)\ 250$$
$$= 3443.86 \times 10^3 \text{ W} = 3443.86 \text{ kW}$$

Propulsive efficiency =

Overall efficiency =

Thermal efficiency,

Example 17.4

A turbo-jet unit is flying at a speed of 268 m/s at an altitude where the ambient conditions are 0.2 bar and

220 K. The air enters an ideal diffuser and leaves the compressor at 1350 K and 1 bar. The fuel supplied has a heating value of 43,000 kJ/kg. Assume all compression and expansion processes to be isentropic. Determine (a) the air-fuel ratio, (b) the specific thrust, and (c) the propulsive efficiency. Take C_{pa} = 1.005 KJ/kg.K and γ_a = 1.4 for compression and C_{pg} = 1.102 KJ/kg, γ_g = 1.33 for expansion.

[IES, 1998]

Figure 17.10 *T-s diagram*

Solution

The *T*-s diagram is shown in Fig. 17.10.

Diffuser:

Compressor:

Combustion chamber:

$$T_4 = 1350 \text{ K}$$
$$h_3 + q_{3-4} = h_4$$
$$\dot{m}_a \, c_{pa} \, T_3 + \dot{m}_f \times \text{CV} = (\dot{m}_a + \dot{m}_f) \, c_{pg} \, T_4$$

Turbine:

$$w_t = w_c$$
$$(\dot{m}_a + \dot{m}_f) \, c_{pg} \, (T_4 - T_5) = \dot{m}_a \times c_{pa} \, (T_3 - T_2)$$
$$37.5 \times 1.102 \, (1350 - T_5) = 36.5 \times 1.005 \, (348.4 - 255.73)$$

or $T_5 = 1267.77$ K

Let $p_6 = 0.2$ bar

Work output = $(\dot{m}_a + \dot{m}_f)\, c_{pg}\, (T_5 - T_6)$

Specific thrust = work output per kg of fuel consumed

$$= 37.5 \times 1.102\ (1267.77 - 905.6)$$
$$= 14966.4 \text{ kN/kg}$$

Jet velocity,

Propulsive efficiency, neglecting fuel mass as compared to air mass,

Example 17.5

A turbojet engine consumes air at the rate of 60 kg/s when flying at a speed of 960 km/h. Calculate the

following:

1. Exit velocity of the jet when the enthalpy drop in the nozzle is 220 kJ/kg and velocity coefficient is 0.95.
2. Fuel rate of flow for A/F ratio of 60:1,
3. Thrust specific fuel consumption (TSFC),
4. Thermal efficiency of the engine assuming combustion efficiency 92% and calorific value of fuel used 43,000 kJ/kg,
5. Propulsive power,
6. Propulsive efficiency, and
7. Overall efficiency

Solution

1. Exit velocity of jet,
2. Fuel rate of flow,
3. Thrust produced, F = $(\dot{m}_a + \dot{m}_f)\, c_j - \dot{m}_a\, c_a$
4. Thermal efficiency
5. Propulsive power
6. Propulsive efficiency
7. Overall efficiency

Example 17.6

The following data refer to turbo-jet flying at a height of 9000 m:

Speed of turbo-jet = 800 km/h

Propulsive efficiency = 54%

Overall efficiency of plant = 18%

Propulsive force = 6000 N

Calorific value of fuel = 44,000 kJ/kg

Calculate (a) absolute velocity of jet, (b) volume of air compressed per main, (c) diameter of jet, (d) power output of the unit, and (e) air-fuel ratio.

Solution

1. Propulsive efficiency,

   Absolute velocity of jet = $c_j - c_a$ =600.76 − 222.2 = 378.56 m/s

2. Propulsive force = $\dot{m_a}\,(c_j - c_a)$

$$6000 = \dot{m_a} \times 378.56$$

$$\dot{m}_a = 5.85 \text{ kg/s}$$

Volume flow rate of air

3. Let jet diameter = d
4. Thrust power,
5. Overall efficiency

Example 17.7

A high altitude flight jet propeller aircraft is flying with speed corresponding to Mach number of 1.25. The ambient atmospheric pressure and temperature are 0.01MPa and –40°C. The temperature and pressure of gases entering the turbine are 827°C and 0.2 MPa. Isentropic efficiency of compressor and turbine are 0.80 and 0.85, respectively. The ram air

efficiency is 0.80. The back pressure on the nozzle may be assumed as the ambient pressure and efficiency of nozzle based on total pressure drop available is 0.90. Neglecting mass increase due to fuel consumed, calculate (a) compressor power per kg per second, (b) air-fuel ratio if the fuel calorific value is 41500 kJ/kg, (c) pressure of gases leaving the turbine, and thrust per kg per second, (d) propulsive efficiency, and (*e*) thermal and overall efficiency. Assume c_{pa} = 1.005 kJ/kgK, γ_a = 1.4, c_{pg} = 1.12 kJ/kgK, γ_a = 1.33.

Solution

Sonic velocity at ambient

conditions,

$$= 0.287 \text{kJ/kg.K}$$
$$T_1 = 273 - 40 = 233 \text{ K}$$

Speed of aircraft,

$$c_a = 1.25 \times 206 = 382.5 \text{ m/s}$$

Ramming process 1-2′ (see Fig. 17.11)

or $T_2 = T_{2'} = 233 \times .3125 = 305.8$ K

Figure 17.11 *T-s diagram for jet propeller*

or $= 0.8 (0.0259 - 0.01) + 0.01 = 0.0227$ MPa

Compressor process 2′–3:

Compressor power, $P_c = m\, c_{pa}\, (T - T)$

$$= 1 \times 1.005\, (635.3 - 305.8) = 331.18 \text{ kJ/kg/s or kW/kg}$$

Combustion process 3′–4:

$$T_4 = 827 + 273 = 1100 \text{ K}$$

Heat supplied, $q_{3'-4} = c_{pg}\, (T_4 - T_{3'}) = 1.12\, (1100 - 635.3) = 520.5$ kJ/kg

$$m_a\, c_{pg}\, (T_4 - T_{3'}) = m_f \times \text{CV}$$

∴ A/F ratio = 79.74:1 = 79.74

Now, $w_c = w_t$

or $\quad 331.18 = c_{pg} (T_4 - T_5') = 1.12 (1100 - T_5')$

or $\quad T_5' = 804.3$ K

$$T_{6'} = T_{5'} - \eta_n (T_{5'} - T_6)$$
$$= 804.3 - 0.9 (804.3 - 559.4) = 583.9 \text{ K}$$
$$(\Delta h)_{\text{nozzle}} = c_{pg} (T_{5'} - T_6) = 1.12 (804.3 - 583.09) = 246.83 \text{ kJ/kg}$$

Thrust $c_{6'} - c_a = 702.6 - 382.5 = 320.1$ kN/kg/s

Propulsive efficiency,

Thermal efficiency,

$$= 0.3337 \text{ or } 33.37\%$$

Overall efficiency $= \eta_{th} \times \eta_p = 0.3337 \times 0.705$

$$= 0.2352 \text{ or } 23.52\%$$

Example 17.8

A flying turbojet engine propels at speed of 880 km/h and draws air at the rate of 50 kg/s. Isentropic enthalpy drop in the nozzle is 190 kJ/kg and its velocity coefficient is 0.96. Fuel air ratio used is 0.012. The calorific value of fuel used is 46000 kJ/kg and combustion efficiency is 95%. Find thermal, propulsive, and overall efficiency.

Solution

Given:

$\Delta h = 190$ kJ/kg, $C_c = 0.96$, F/A = 0.012, CV = 46000 kJ/kg, $\eta_{\text{comb}} = 0.95$,

1. Thermal efficiency,
2. Propulsive efficiency,
3. Overall efficiency,

Example 17.9

The effective jet exit velocity from jet engine is 2700 m/s. The forward flight velocity is 1350 m/s and the air flow rate is 78.6 kg/s. Calculate (*a*) thrust, (*b*) thrust power, and (*c*) propulsive efficiency.

Solution

Given: $c_j = 2700$ m/s, $c_a = 1350$ m/s, $\dot{m}_a = 78.6$ kg/s

1. Thrust, F = $\dot{m}_a\,(c_j - c_a)$ = 78.6(2700 – 1350) = 106110N
2. Thrust power,
3. Propulsive power P.P

Propulsive efficiency

Example 17.10

A simple turbo-jet unit operates with a turbine inlet temperature of 1100 K, a pressure ratio of 4:1 and mass flow of 22.7 kg/s under design conditions. The following component efficiencies may be assumed

Isentropic compressor efficiency = 0.85

Isentropic turbine efficiency = 0.9

Propeller nozzle efficiency = 0.95

Transmission efficiency = 0.99

Combustion chamber pressure loss = 0.21 bar

Calculate the design thrust and specific fuel consumption when the unit is stationary at sea level where the conditions may be taken as 1.013 bar and 288 K.

Solution

Given that $T_3 = 1100$ K, $r_p = 4$, $\dot{m}_a = 22.7$ kg/s, $\eta_c = 0.85$, $\eta_t = 0.9$, $\eta_n = 0.95$, $\eta_{\text{mech}} = 0.99$, $\Delta p_3 = 0.21$ bar, $p_1 = 1.013$ bar, $T_1 = 288$ K

Process 1 – 2: Compressor (Fig. 17.12)

or $T_{2'} = 452.67$ K

$$p_{3'} = p_3 - \Delta p_3 = 4 \times 1.013 - 0.21 = 3.842 \text{ bar}$$

Neglecting the mass of fuel and $c_{pg} = c_{pa}$

$$(T_3 - T_{4'})\eta_{\text{mech}} = T_{2'} - T_1$$

or $(1100 - T_{4'}) \times 0.99 = 452.67 - 288$

or $T_{4'} = 933.67$ K

Figure 17.12 *T-s diagram for simple turbo-jet*

Taking $p_0 = p_1$

or $T_{5'} = 775.13$ K

Enthalpy drops per kg of mass flow rate per second:

$$(\Delta h)_c = c_p\,(T_{2'} - T_1)$$
$$(\Delta h)_t = c_p\,(T_3 - T_{4'})$$
$$(\Delta h)_n = c_p\,(T_{4'} - T_{5'})$$
$$\Delta h\ [(\Delta h)_t + (\Delta h)_n] - (\Delta h)_c = c_p[T_3 - T_{4'} + T_{4'} - T_{5'} - T_{2'} + T_1]$$
$$= c_p[T_3 - T_{5'} - T_{2'} - T_1] = 1.005[1100 - 775.13 - 452.67 + 288]$$
$$= 161 \text{ kJ/kg}$$

Velocity of nozzle,

1. Design thrust, F = $\dot{m}_a \times c_j$ = 22.7 × 567.45 = 12881 N
2. Specific fuel consumption

Example 17.11

In a turbo jet engine, air enters the diffuser at 0.8 bar, 240 K. with a velocity of 1000 km/hr. The pressure, ratio across the

compressor is 8. The turbine inlet temperature is 1200 K and the pressure at nozzle exit is 0.8 bar. The turbine work just equals the compressor work input. The diffuser, compressor, turbine and nozzle processes are isentropic and there is no pressure drop for flow through the combustor. Determine the pressure at exit from the diffuser, the compressor, and the turbine, and also the velocity at the nozzle exit. Show the various process on a temperature-entropy diagram. For air, c_p 1.001 kJ/kg. K and $c_p/c_v = 1.4$.

[IES, 1993]

Solution

Given that $p_1 = 0.8$ bar, $T_1 = 240$ K, $c_1 = 1000$ km/h or 277.78 m/s, $T_4 = 1200$ K, $w_t = w_c$, $c_{pa} = 1.001$ kJ/kg K, $\gamma_a = 1.4$

From Fig. 17.13 we have

$$p_3 = p_4,\ p_1 = p_6$$

Assume $c_{pg} = c_{pa}$ and $\gamma_g = \gamma_a = \gamma$

Diffuser:

Compressor:

$$p_3 = 8 \times 1.34 = 10.778 \text{ bar}$$
$$p_4 = p_3 = 10.778 \text{ bar}$$

Figure 17.13 *Turbo-jet engine T-s diagram*

Turbine:

$$w_c = c_p\,(T_3 - T_2) = 1.001\,(504.5 - 278.5) = 226.226 \text{ kJ/kg}$$
$$w_t = c_p(T_4 - T_5) = 1.001\,(1200 - T_5)$$

Now $\quad w_t = w_c$

$$\therefore 1.001\,(1200 - T_5) = 226.226$$

or $\quad T_5 = 974 \text{ K}$

Nozzle:

Nozzle jet velocity

Example 17.12

A jet engine is flying at 300 m/s when the pressure and temperature

of the atmosphere are 0.8 bar and 230 K, respectively. The compressor pressure ratio is 4 and the maximum cycle temperature is 1000 K. Calculate, specific thrust, power produced, propulsive efficiency, overall thermal efficiency, and the fuel consumption. Assume: isentropic efficiency of components unity; nozzle throat area 0.06 m^2; calorific value of fuel 43000 kJ/kg; c_p and γ for the combustion and expansion processes 1.15 kJ/kg K and 1.333.

[IES, 1997]

Solution

Given: $c_1 = 300$ m/s, $p_1 = 0.8$ bar, $T_1 = 230$ K, $p_3/p_2 = 4$, $T_4 = 1000$ K,

$\eta_{\text{isen}} = 1.0$, $A_t = 0.06\ \text{m}^2$, C.V. = 43000 kJ/kg

$c_{pa} = 1.005$ kJ/kg. K, $\gamma_a = 1.4$, $c_{pg} = 1.15$ kJ/kg. K, $\gamma_g = 1.333$

The *T–s* diagram is shown in Fig. 17.14.

Diffuser:

Figure 17.14 *T-s diagram for jet engine*

Compressor:

$$p_3 = 4p_2 = 4 \times 1.49 = 5.96 \text{ bar}$$
$$T_4 = 1000 \text{ K}$$

Neglecting mass of fuel as compared to that of air,

Work done by turbine = Work done on compressor

$$c_{pg}\,(T_4 - T_5) = c_{pa}\,(T_3 - T_2)$$

$$1.15\,(1000 - T_5) = 1.005\,(408.32 - 274.78)$$

or $\quad T_5 = 883.3$ K

Also $p_4 = p_3$

Let $p_6 = p_1 = 0.8$ bar

Enthalpy drop in nozzle,

$h = c_{pg} (T_5 - T_6) = 1.15 (883.3 - 605.4) = 319.6$ kJ/kg

Velocity of jet,

Thrust produced, $F = \dot{m}(c_j - c_a) = 1(799.5 - 800) = 499.5$ N/kg of air

Thrust power,

Specific thrust = 149.85 kW/kg of air

Power produced = 149.85 + Turbine output

= 149.85 + 1.15 (1000 – 883.3) = 284.06 kW/kg of air

Propulsive efficiency,

Overall thermal efficiency,

1. b. Considering mass of fuel.

Let m_f = mass of fuel.

Then $m_f \times \text{CV} = (m_a + m_f)\, c_{pg} \times T_4 - c_{pa}\, m_a\, T_3$

Now $W_t = W_c$

$$(m_a + m_f)\, c_{pg}\, (T_4 - T_5) = m_a\, c_{pa}\, (T_3 - T_2)$$

or $\quad (1 + 0.0176) \times 1.15\, (1000 - T_5) = 1.005$ (408.32 – 274.78)

or $\quad T_5 = 885.3$ K

Nozzle:

$\Delta h = c_{pg}\, (T_5 - T_6) = 1.15\, (885.3 - 604.98) = 322.37$ kJ/kg

Specific thrust,

$= (1 + 0.0176)\ (802.95 - 300) = 511.8$ N/kg of air

Now

$p_6 v_6 = RT_6$

Mass flow rate,

Power produced $= F \times c_a \times \dot{m}/10^3 = 511.8 \times 300 \times 22.2/10^3 = 3408.6$ kW

Example 17.13

Air at 25 kPa and 230 K enters a turbojet engine with a velocity of 250 m/s. The pressure ratio across the compressor is 12. The turbine inlet temperature is 1400 K and the pressure at the nozzle exit is 25 kPa. The diffuser, compressor, turbine and nozzle processes are isentropic and there is no pressure drop for flow through the combustor. Draw the line diagram indicating all the components and show the processes on a *T*-*s* diagram. Under steady state operating conditions, determine (a) the velocity at the nozzle exit and (b) the pressures and temperatures at each state. **[IES, 2006]**

Solution

Diffuser - compressor: The line diagram of turbo-jet and the *T-S* diagram are shown in Fig. 17.15.

Figure 17.15 *Turbo-jet engine: (a) Line diagram of turbo-jet, (b) T-s diagram*

Combustor:

$$p_3 = p_2 = 467.64 \text{ kPa}$$
$$T_3 = 1400 \text{ K}$$

Turbine: Assuming compressor work = turbine work, we have

$$c_p (T_2 - T_1) = c_p (T_3 - T_4)$$

or $\quad 531.05 - 262.1 = 1400 - T_4$

or $\quad T_4 = 1130.05$ K

Nozzle:

$c_4 \approx 0$ as compared to c_5.

$$c_5 = [2\,(h_4 - h_5)]^{0.5} = [2 \times c_p\,(T_4 - T_5)]^{0.5}$$
$$= (2 \times 1000)^{0.5} \times [1.005\,(1130.5 - 606.3)]^{0.5}$$
$$= 1026 \text{ m/s}$$

Summary for Quick Revision

1. In jet propulsion, the work output of a gas turbine is used to produce high velocity jet by expanding gas through a nozzle. The reaction of the jet produces the propulsive force to propel a vehicle.
2. Jet propulsion systems are of two types: air stream jet engines and self-contained rocket engines.
3. In a screw propeller, full expansion of gas takes place in the turbine and the total power developed is used to drive the compressor and the propeller.
4. In the ramjet engine, the diff users convert the kinetic energy of high speed entering air into pressure rise (called ram effect). The high pressure and high temperature gases are then passed through the nozzle which converts the pressure energy into kinetic energy to produce a high velocity jet.

5. The pulse jet engine develops thrust by a high velocity jet of exhaust gases without the aid of a compressor or turbine.
6. The turbine of a turbo-jet engine produces just sufficient power to drive the compressor by partial expansion in the turbine. The gases coming out from the turbine are expanded in the nozzle to produce a very high velocity jet to propel the aircraft.
7. In the turbo-prop engine, the gases are expanded partly in the turbine and partly in the nozzle. The turbine power is totally consumed to drive the compressor and the propeller. The propeller provides most of the propulsive thrust. It is suitable for pilotless operation of aircraft.
8. In a rocket engine, both the fuel and oxidiser are stored in the propelling body. It is the only device capable to space flight.
9. Turbo-jet engine:
    1. Thrust, $F = \dot{m}(c_j - c_a)$N/(kg/s) of air

       where c_j, c_a = velocity of jet and aircraft, respectively

       $$\dot{m} = 1 + \dot{m}_f/\dot{m}_a$$

       $\dot{m}_f$, $\dot{m}_a$ = mass of fuel and air, respectively
    2. Thrust Power, T.P. =
    3. Propulsive power, P.P
    4. Propulsive efficiency,
    5. Thermal efficiency,
    6. Overall efficiency,
    7. For η_o to be maximum,
    8. Jet efficiency,
    9. Ram air efficiency,

       where Mach number,

       Sonic velocity,
10. Rocket engine:
    1. Thrust work = $c_j\ c_a$
    2. Propulsive work
    3. Propulsive efficiency

Multiple-choice Questions

1. In solid propellants rocket, ammonium picrate is usually added as:
    1. an additive
    2. an inhibitor
    3. a darkening agent
    4. a plasticiser
2. A turbo prop is preferred to turbo-jet because
    1. it has high propulsive efficiency at high speed
    2. it can fly at supersonic speed
    3. it can fly at high elevations
    4. it has high power for take-off
3. The thrust of a jet propulsion power unit can be increased by
    1. injecting water into compressor
    2. burning fuel after gas turbine
    3. injecting ammonia into the combustion chamber
    4. All of the above
4. Figure 17.16 shows the propulsive efficiencies of three different engines. Based on the figure, match List I with List II and select the correct answer using the codes given below the List:

    Codes:

    A B C

    1. 1 2 3
    2. 2 1 3
    3. 2 3 2
    4. 3 1 2

Figure 17.16

5. If V is the jet velocity V is the vehicle velocity, the propulsive efficiency of a rocket is given by
    1.
    2.
    3.
    4.
6. Consider the following statements relating to rocket engines:
    1. The combustion chamber in a rocket engine is

directly analogous to the reservoir of a supersonic wind tunnel.
2. Stagnation conditions exist at the combustion chamber.
3. The exit velocities of exhaust gases are much higher than those in jet engines.
4. Efficiency of socket engines is higher than that of jet engines.

Of these statements:

1. I, III, and IV are correct
2. II, III, and IV are correct
3. I, II, and III are correct
4. I, II, and IV are correct

7. Consider the following statements about a rocket engine:
    1. It is a very simple in construction and operation.
    2. It can attain very high vehicle velocity.
    3. It can operate or very long duration.

Of these statements:

1. I and II are correct
2. I and III are correct
3. II and III are correct
4. I, II, and III are correct

8. Only rocket engines can be propelled to space because
    1. they can generate very high thrust.
    2. they have high propulsion efficiency.
    3. these engines can work on several fuels.
    4. they not air-breathing engines.
9. Consider the following statements:

As compared to turboprop, a turbojet

1. can operate at higher altitudes
2. can operate at higher flight velocities
3. is more fuel efficient at lower speeds

Of these statements:

1. I, II, and III are correct

2. I and II are correct
3. II and III are correct
4. I and III are correct

10. Propulsion efficiency of a jet engine is given by (where u is flight velocity and V is jet velocity relative to aircraft).
    1. $2u\ [V - u]$
    2. $[V + u]\ 2u]$
    3. $2u\ [V + u]$
    4. $[V - u]\ 2u]$
11. Consider the following statements:

    In open cycle turbo-jet engines used in military aircraft, reheating the exhaust gas from the turbine by burning more fuel is used to increase

    1. thrust
    2. the efficiency of engine
    3. the range of aircraft

    Of these statements:

    1. I and III are correct
    2. I and II are correct
    3. II and III are correct
    4. I, II, and III are correct
12. In a turbojet engine, subsequent to heat addition to compressed air, to get the power output, the working substance is expanded in
    1. turbine blades, which is essentially an isentropic process
    2. turbine blades, which is polytropic process
    3. exit nozzle, which is essentially an isentropic process
    4. exit nozzle, which is a constant volume process
13. Which one of the following is the correct sequence of the position of the given components in a turboprop?
    1. Propeller, compressor, turbine, burner
    2. Compressor, propeller, burner, turbine
    3. Propeller, compressor, burner, turbine
    4. Compressor, propeller, turbine, burner
14. The absolute jet exit velocity from a jet engine is 2800 m/s and the forward flight velocity is 1400 m/s. The propulsive efficiency is

1. 33.33%
2. 40%
3. 66.67%
4. 90%

15. The efficiency of jet engine is
    1. higher at high speed
    2. lower at low speed
    3. higher at high altitude
    4. same at all altitude
16. The propulsive efficiency of a turbojet aircraft approaches 100% when the thrust approaches
    1. Maximum
    2. 50% of the maximum
    3. 25% of the maximum
    4. Zero
17. Which one of the following is the correct expression for the propulsive efficiency of a jet plane (neglecting the mass of fuel)?
    1.
    2.
    3.
    4.
18. The relative jet exit velocity from a rocket is 2700 m/s. The forward flight velocity is 1350 m/s. What is the propulsive efficiency of the unit?
    1. 90%
    2. 66.66%
    3. 50%
    4. 33.33%
19. Consider the following statements regarding performance of turbojet engines:
    1. The thrust decreases at higher altitudes due to reduced density of air and consequently lower mass flow of air.
    2. Any subsonic speed, the effect of increased velocity is to increase the air flow and the thrust increases.
    3. The relative velocity of jet with respect to the medium decreases at higher speeds which tends to reduce the thrust.
    4. For turbojet engine, the thrust of jet at subsonic speeds remains relatively constant. Which of the statements given above are correct?
    1. I, II, III, and IV

2. I and III
3. I, II, and IV
4. II, III, and IV

20. Consider the following statements:

The thrust of rocket engine depends on

1. Effective jet velocity
2. Weight of the rocket
3. Rate of propellant consumption

Of these statements:

1. I and II are correct
2. I and III are correct
3. II and III are correct
4. I, II, and III are correct

21. Consider the following statements:

In a turbojet engine, thrust may be increased by

1. Increasing the jet velocity
2. Increasing the mass flow rate of air
3. After burning of the fuel

Of these statements:

1. I and II are correct
2. II and III are correct
3. I and III are correct
4. I, II, and III are correct

22. The propulsive efficiency of a turbojet aircraft approaches 100% when the thrust approaches
    1. maximum
    2. 50% of the maximum
    3. 25% of the maximum
    4. zero
23. An aeroplane travels at 400 km/h at sea level where the temperature is 15°C. The velocity of the aeroplane at the same Mach number at an altitude where a temperature of –25°C is prevailing, would be
    1. 126.78 km/h

   2. 130.6 km/h
   3. 371.2 km/h
   4. 400.10 km/h
24. In turbo prop, the expansion of gases takes place approximately
    1. 100% in the turbine
    2. 80% in the turbine and 20% in the nozzle
    3. 50% in the turbine and 50% in the nozzle
    4. 100% in the nozzle
25. With reference to turbojet and rocket engines, consider the following statements:
    1. Efficiency of rocket engines is higher than that of jet engines.
    2. Exit velocities of exhaust gases in rocket engines are much higher than those in jet engines.
    3. Stagnation conditions exist at the combustion chamber in rocket engines.
    4. Rocket engines are air-breathing engines.

    Which of these statements are correct?

    1. I and II
    2. I, III, and IV
    3. II, III, and IV
    4. I, II, and III
26. Consider the following statements indicating a comparison between rocket and jet propulsion systems:
    1. Both rocket and jet engines carry the fuel and oxidant
    2. Rockets do not employ compressor or propeller
    3. Rockers can operate in the vacuum too
    4. Rockers can use solid fuels and oxidants
    1. I, II, III, and IV
    2. Only I and II
    3. Only II, III, and IV
    4. Only I, III, and IV
27. Match List I with List II in respect of a chemical rocket engine and select the correct answer using the codes given below the Lists:

    Codes:

	A	B	C	D
1.	1	3	2	4
2.	2	3	4	1
3.	2	4	3	1
4.	4	1	2	3

Explanatory Notes

1. 14. (c)
2. 18. (b)
3. 22. (c)
4. 23. (c)

Review Questions

1. Explain the principle of jet propulsion.
2. List the various jet propulsion systems.
3. Which are piloted and pilotless jet propulsion systems?
4. How is rocket propulsion different from jet propulsion?
5. Describe the working of a screw propeller. What are the advantages of ramjet engine?
6. Explain ram effect and ram pressure. What are the advantages of ram-ret engine?
7. Explain the working of a pulse-jet engine. What are its limitations?
8. Describe the construction of a turbo-jet engine. What are its advantages and disadvantages?
9. How is a turbo-prop engine different from a turbo-jet engine?
10. Define thrust and thrust power.
11. Define propulsive power and propulsive efficiency.
12. Define thrust specific fuel consumption (TSFC).
13. Define thermal efficiency and overall efficiency.
14. What is jet efficiency and nozzle efficiency?
15. Define thrust work, propulsive work, and propulsive efficiency for a rocket engine.

Exercises

17.1 In a propulsion unit, air is drawn into the compressor at 15°C, 1 bar, and delivered at 4 bar. The isentropic efficiency of compression is 82% and the compression is uncooled. The air is then heated at constant pressure to 750°C. The air then passes through a turbine which drives the compressor only and has an isentropic efficiency of 78%, followed by expanding through a nozzle with an efficiency of 88% to atmospheric pressure of 1 bar. Neglecting any mass increase due to the mass of fuel, determine: (a) the power required to drive the compressor, (b) the air-fuel ratio if calorific value of fuel is 42,000 kJ/kg (c) pressure of gases leaving the turbine, and (d) thrust per kg of air per second. Assume for air: R =

0.287 kJ/kg K, $\gamma = 1.4$.

[**Ans.** 15.1 171.38 kW, 74.1:1, 1.72 bar, 469.95 N/kg/s]

17.2 A jet propelled plane consuming air at the rate of 18.2 kg/s is to fly at Mach number 0.6 of an altitude of 4.5 km ($p_a = 0.55$ bar, $T_a = 255$ K). The diffuser which has a pressure ratio of 5 and maximum temperature in the combustion chamber is 1273 K. After expanding in the turbine the gases continue to expand in the nozzle to a pressure of 0.69 bar. The isentropic efficiencies of compressor, turbine, and nozzle are 0.81, 0.85, and 0.915, respectively. The heating value of fuel is 45870 kJ/kg. Given $c_{pa} = 1.005$ kJ/kg.K, $c_{pg} = 1.147$ kJ/kg.K, $\gamma_a = 1.4$, $\gamma_g = 1.33$. Calculate (a) power input to the

compressor, (b) power output of the turbine, (c) fuel air ratio, (d) thrust provided by the engine, and (e) thrust power developed.

[**Ans.** 3608.27 kW, 3608.27 kW, 0.0205, 9087 N, 1743.96 kW]

17.3 A turbo-prop aircraft is flying at 800km/h at an altitude, where the ambient conditions are 0.567 bar and –20°C. Compressor pressure ratio is 8:1. Maximum gas temperature is 1100 K. The intake duct efficiency is 0.95 and total head isentropic efficiency of compressor and turbine is 0.92 and 0.95, respectively. Calculate the specific power output in kJ/kg.s air/s, the thermal efficiency of the unit taking mechanical efficiency of transmission as 0.96. Assume that exhaust gases leave

the aircraft at 800 km/h relative to the aircraft. Take $c_{pa} = 1.005$ kJ/kg.K, $\gamma_a = 1.4$, $c_{pg} = 1.147$ kJ/kg. K, $\gamma_g = 1.33$.

[**Ans.** 226.23 kJ/kg, 34.17%]

17.4 In a jet-propulsion unit air is compressed by means of uncooled rotary compressor, the pressure at the delivery being 3.5 times of that at the entrance, and the temperature rise during compression is 1.15 times of that for frictionless adiabatic compression. The air is then led to the combustion chamber where the fuel is burned under constant pressure conditions. The products of combustion at 480°C pass through turbine which drives the compressor. The exhaust gases form the turbine are expanded in the nozzle down

to atmospheric pressure, the ambient conditions are 1 bar, 10°C. Calculate the power required to drive the compressor per kg of air per second, the air-fuel ratio if the calorific value of fuel is 43300 kJ/kg, and static thrust developed per kg of air per second. Assume that the values of R and g after combustion remain same as that for air. R = 0.287 kJ/kg.K, $\gamma = 1.4$.

[**Ans.** 140.92 kJ/kg/s, 130.74:1, 414.93 N/kg of air/s]

17.5 The speed of a jet-propulsion aircraft is 200 m/s relative to still air. The total pressure and total temperature at intake to uncooled compressor are 0.7 bar and 0°C. The total temperature and total pressure of gases entering the turbine are 760°C and 3.15 bar. The

isentropic efficiencies of compressor and turbine are 85% and 80%, respectively. The static back pressure on the propulsion nozzle is 0.55 bar and the efficiency of nozzle based on the total pressure drop available is 0.90%.

Neglecting other losses and mass increases due to fuel consumed, determine (a) the power required to drive the compressor per kg per second, (b) the air-fuel ratio of the fuel has calorific value of 41860 kJ/kg, (c) the total pressure of gases leaving the turbine, and (d) the thrust per kg per second. Assume C_{pa} = 1.005 kJ/kg. K, γ_a = 1.4., C_{pg}= 1.17 kJ/kg. K, and γ_g = 1.33.

[**Ans.** 173.65 kW/kg, 63.1:1, 1.418 bar, 426 N/kg.s]

17.6 A turbojet aircraft is flying at a speed such that the corresponding Mach number is 0./9, where the ambient conditions are 0.5 bar and -20°C. The compressor pressure ratio is 8:1. The maximum cycle temperature is 1250 K with fuel of calorific value 44000 kJ/kg. The pressure loss in the combustor is 0.1 bar. The ram air efficiency = 0.9, isentropic efficiency of compressor = 0.85, isentropic efficiency of turbine 0.80, combustion efficiency = 0.98, transmission efficiency of turbine to compressor = 0.98, nozzle efficiency = 0.90, and outlet area of nozzle = 0.1 m. Calculate mass flow rate, the thrust developed, and specific fuel consumption. Assume c_{pa} = 1.004 kJ/kg. K, γ_a = 1.4, c_{pg} = 1.15kJ/kg. K, and γ_g =

1.335.

[**Ans.** 19.25 kg/s, 147.5 kJ/kg, 0.04 hg/kwh]

17.7 The exit velocity from a jet unit is 650 m/s for an air flow of 40hg/s through the unit. The aircraft is flying at 250 km/h. Calculate the thrust developed, the thrust power and the propulsion efficiency. Neglect the effect of fuel.

[**Ans.**23222.4 N, 1612.56 kW, 193%]

ANSWERS TO MULTIPLE-CHOICE QUESTIONS

1. a
2. d
3. a
4. a
5. a
6. c
7. b
8. d
9. b
10. c
11. b
12. a
13. c
14. c
15. a

16. d
17. b
18. b
19. b
20. b
21. a
22. d
23. c
24. a
25. a
26. c
27. c

Chapter 18

Introduction to Refrigeration

18.1 ❐ INTRODUCTION

The process of producing cold by artificial means may be called refrigeration. The term ‘*refrigeration*’ may be defined as the artificial withdrawal of heat, producing in a substance or within a space, a temperature lower than that which would exist under the natural influence of surroundings. In other words, the refrigeration means a continued extraction of heat from a body whose temperature is already below the temperature of its surroundings. In

refrigeration, heat is virtually being pumped from a lower temperature to a higher temperature.

According to the second law of thermodynamics, the extraction of heat from a body at lower temperature to a higher temperature can only be performed with the aid of some external device. Such a device is called *refrigerator*. A refrigerator is a reversed heat engine or a heat pump which extracts heat from a cold body and delivers it to a hot body. The substance or medium which works in a heat pump to extract heat from a cold body and to deliver to a hot body is called a *refrigerant*.

18.2 ❐ REFRIGERATION SYSTEMS

A refrigeration system is a device to produce the refrigeration effect. Some of the popular refrigeration systems are:

1. Ice refrigeration-used in hotels for keeping the drinks cold.
2. Evaporative refrigeration-desert bag, artificial snow.
3. Air refrigeration-aircrafts.
4. Gas throttling refrigeration-liquefaction of gases, like air, nitrogen and oxygen.
5. Vapour compression refrigeration-domestic refrigerator.
6. Vapour absorption refrigeration-ice making using aqua ammonia or water plus lithium bromide as the absorbent.
7. Steam jet refrigeration-comfort cooling in air-conditioning installations or in industrial processes.
8. Liquid gases refrigeration-transportation vehicles for perishable items.
9. Dry ice (solid CO_2) refrigeration-transportation of perishable items.

18.3 ❐ METHODS OF REFRIGERATION

The methods of refrigeration are:

1. Vapour compression refrigeration.
2. Vapour absorption refrigeration.
3. Ejector-compression refrigeration.
4. Electro-Lux refrigeration.
5. Solar refrigeration.
6. Thermo-electric refrigeration.
7. Vortex-tube refrigeration.

18.3.1 Vapour Compression Refrigeration System

The schematic diagram of simple vapour compression refrigeration system is shown in Fig. 18.1.

It consists of the following five essential parts:

1. **Compressor:** The low pressure and temperature vapour refrigerant from evaporator is drawn into the compressor through the inlet or suction valve *A*, where it is compressed to a high pressure and temperature. This high pressure and temperature vapour refrigerant is discharged into the condenser though the delivery, or discharge valve *B*.
2. **Condenser:** The condenser or cooler consists of coils of pipe in which the high pressure and temperature vapour refrigerant is cooled and condensed. The refrigerant, while passing through the condenser, gives up its latent heat to the surrounding condensing medium which is normally air or water.

Figure 18.1 *Simple vapour compression refrigeration system*

3. **Receiver:** The condensed liquid refrigerant from the condenser is stored in a vessel known as receiver from where it is supplied to the evaporator through the expansion valve or refrigerant control valve.
4. **Expansion valve:** It is also called throttle valve or refrigerant control valve. The function of the expansion valve is to allow the liquid refrigerant under high pressure and temperature to pass at a controlled rate after reducing its pressure and temperature. Some of the liquid refrigerant evaporates as it passes through the expansion valve, but the greater portion is vapourised in the evaporator at the low pressure and temperature.
5. **Evaporator:** An evaporator consists of coils of pipe in which the liquid vapour refrigerant at low pressure and temperature is

evaporated and changed into vapour refrigerant at low pressure and temperature. In evaporating, the liquid vapour refrigerant absorbs its latent heat of vaporisation from the medium (air, water or brine) which is to be cooled.

Note: In any compression refrigeration system, there are two different pressure conditions. One is called the *high pressure side* and other is known as *low pressure side*. The high pressure side includes the discharge line (i.e. piping from delivery valve *B* to the condenser), condenser, receiver and expansion valve. The low pressure side includes the evaporator, piping from the expansion valve to the evaporator and the suction line (i.e. piping form the evaporator to the suction valve *A*).

18.3.2 Vapour Absorption System

A simple vapour absorption system consists of a condenser, an expansion

device, an evaporator, an absorber, a pump, a generator and a pressure reducing valve. The system can be improved by incorporating a regenerative heat exchanger between the poor and rich solutions. The two commonly used refrigerant absorbent pairs are ammonia-water (aqua ammonia) and lithium bromide-water. Lithium bromide-water system is superior to ammonia water system. In aqua ammonia, ammonia works as the refrigerant and water as the absorber, whereas in the other pair, water acts as the refrigerant and lithium bromide as the absorber.

A schematic diagram of absorption refrigeration system is shown in Fig.

18.2. The working of the ammonia absorption system is described below:

Ammonia vapour is produced in the generator at high pressure from aqua ammonia by external heating. The water vapour are removed by passing through an analyser and rectifier. The dehydrated ammonia enters the condenser where vapours are condensed. The sub-cooled high pressure liquid ammonia is passed through a throttle valve to further lower its temperature. Now they enter the evaporator and leave as saturated vapour. The saturated vapour goes to absorber to become strong after absorbing ammonia vapour and the strong solution is pumped to the generator through the pump and heat

exchanger. The solution in the generator becomes weak as ammonia vapour comes out of it. The weak ammonia solution from the generator is passed to the heat exchanger through the pressure reducing valve. The cycle is repeated.

18.3.3 Ejector-Compression System

The system using water as the refrigerant in a vapour compression cycle in which compression is achieved by the principle of jet compression employing its own vapour as the motive vapour is called *steam-ejector system*. Refrigeration effect is obtained by direct evaporation and subsequent self cooling of water.

Figure 18.2 *Ammonia absorption system*

Bubble point temperature is the temperature at which a liquid mixture begins to boil. Azeotrope is a constant temperature boiling mixture.

18.3.4 Electro-Lux Refrigeration

This vapour absorption system uses ammonia as the refrigerant and hydrogen as the inert gas medium. The inert gas is confined only to the low side of the system, i.e., evaporator and absorber. In the evaporator, ammonia evaporates in the presence of hydrogen atmosphere. The partial pressure of ammonia is much lower than its pressure and thus give very low temperature. Hydrogen has the advantage of being non-corrosive and insoluble in water. This system is used

for domestic refrigeration.

18.3.5 Solar Refrigeration

This uses the strong aqua ammonia vapour absorption system by utilizing the solar energy as the heat source to produce the refrigeration effect. This is an intermittent cycle. Lithium chloride-water system has also been used.

18.3.6 Thermo-electric Refrigeration

It is based on the thermoelectric effects: Seebeck effect, Peltier effect, and the Thomson effect. These effects are reversible in nature. The other two irreversible effects associated with thermoelectricity are the energy. Joule effect and conduction effect. It has low COP but infinite life expectancy, no

leakage problem.

18.3.7 Vortex Tube Refrigeration

Vortex tube is a simple device for producing cold. It consists of a nozzle, diaphragm, valve, hot air side and cold air side. This is the only device to produce the refrigeration effect by utilizing the waste heat energy.

18.4 ❐ UNIT OF REFRIGERATION

The capacity of a refrigeration unit is given in tons of refrigeration. A ton of refrigeration is defined as the quantity of heat to be removed in order to form one ton of ice in 24 hours when the initial temperature of water is 0°C. 1 ton of refrigeration, TR = 211 kJ/min or 3.5167 kW. Taking latent heat of freezing of water equal to 336 kJ/kg, we have

Coefficient of performance (COP): The COP of a refrigeration system is a ratio of the heat removed (cooling effect produced) from a system to the work supplied to achieve the heat removal.

where P = energy required in kW

T_n = capacity of plant in tons

18.5 ❐ REFRIGERATION EFFECT

During evaporation, the liquid-vapour refrigerant absorbs its latent heat of vapourisation from the medium (air, water, brine) which is to be cooled. This heat which is absorbed by the refrigerant is called the refrigeration effect. In other words, the refrigeration effect is the

amount of cooling produced.

Refrigeration effect = Enthalpy of vapour refrigerant – Enthalpy of liquid refrigerant.

18.6 ❐ CARNOT REFRIGERATION CYCLE

The reversed Carnot cycle is shown in Fig. 18.3 on *p-v* and *T-s*. diagrams. The sequence of operations is given below:

1. The process 1-2 represents the isentropic compression of the working fluid with the aid of external work. The temperature of the working fluid rises from T_1 to T_2.
2. The process 2-3 represents the isothermal compression of the working fluid during which heat Q_r is rejected at constant high temperature T_2.
3. The process 3-4 represents the isentropic expansion of the working fluid. The temperature of working fluid drops from T_2 to T_1.
4. The process 4-1 represents the absorption of heat Q_a by the working fluid from refrigerator at constant low temperature T_1 during isothermal expansion.

The Carnot cycle is practically not feasible since isothermal energy rejection requires extremely slow

motion followed by isentropic process during which the piston should move at extremely faster rate, which is mechanically not obtainable.

The COP of reversed Carnot cycle is the highest of all refrigeration cycles. COP increases as T_1 is increased or T_2 is lowered. The increase in COP for increased $T_1$is more than for decreased T_2. Since T_2 in winter is less than T_2 in summer therefore COP in winter is more than that of in summer.

If the cycle is used as a heat pump, then energy performance ratio,

Figure 18.3 *Reversed Carnot cycle: (a) p-v diagram, (b) T-s diagram*

18.7 ❒ DIFFERENCE BETWEEN A HEAT ENGINE, REFRIGERATOR AND HEAT PUMP

Heat Engine: In a heat engine, as shown in Fig. 18.4(a), the heat supplied to the engine is converted into useful work. If Q_2 is the heat supplied to the engine and Q_1 is the heat rejected from the engine, then the net work done by the engine is given by,

$$W_e = Q_2 - Q_1$$

Efficiency of heat engine,

Refrigerator: A refrigerator is an equipment used to remove the heat continuously from the space (sink). It maintains the temperature below atmospheric temperature and reject the heat to the atmosphere (source) by

increasing the temperature potential of the heat to be rejected with the help of mechanical work input (compressor).

The refrigeration system is shown in Fig. 18.4(b). In a refrigerator system Q_1 quantity of heat is removed from the source where its temperature T_1 is maintained below atmospheric temperature T_2. COP of refrigerator is,

Heat Pump: It is an equipment used to supply the heat continuously to the space. It maintains the temperature above atmospheric temperature by absorbing the heat from the atmosphere and increasing its temperature potential with mechanical work input (compressor). The heat pump system is

shown in Fig. 18.4(c). In a heat pump system, Q_2 quantity of heat is supplied to the room where the temperature T_2 is maintained above atmospheric temperature T_1. The heat Q_1 is obtained from the atmosphere.

Figure 18.4 *(a) Heat engine, $T_1 > T>_{atm}$, (b) Refridgerator $T_1 < T_{atm}$, (c) Heat pump*

COP or Energy performance ratio of heat pump is,

Example 18.1

Find the COP of a refrigeration system if the work input is 10 kJ/kg and refrigeration effect produced is 180 kJ/kg of refrigerant flowing.

Solution

w_{ref} =100 kJ/kg, q=180 kJ/kg

Example 18.2

A machine working on a Carnot cycle operates between 310 K and 270 K. Calculate the COP when it is operated as:

1. a refrigerator, and
2. a heat pump.

Solution

Given: $T_1 = 270$ K, $T_2 = 310$ K

1. Refrigerator
2. Heat pump:

Example 18.3

A Carnot refrigeration cycle absorbs heat at 268 K and rejects it at 298 K.

1. Calculate the COP of this refrigeration cycle.
2. If the cycle is absorbing 1150 kJ/min at 268 K, find the rate of work input required.
3. How many kJ/s will be delivered if it works as a heat pump at 298 K and absorbs 1150 kJ/min at 268 K.

Solution

Given: $T_1 = 268$ K, $T_2 = 298$ K, $Q_1 = 1150$ kJ/min

1. Carnot refrigeration cycle:

   Work required,

2. Heat pump:

$$(\text{COP})_{HP} = 1 + (\text{COP})_{\text{ref}} = 1 + 8.933 = 9.933$$

18.8 ❐ POWER CONSUMPTION OF A REFRIGERATING MACHINE

The power consumption of a refrigerating machine is determined in terms of kW. However the power

consumption of the driving motor is sometimes rated in horsepower (H.P). we have

where W_{ref} is in kW and HP is in metric units,

where Q_1 = refrigerating capacity in kW

Since 1 TR = 3.5167 kW, therefore

Example 18.4

The ambient air temperature during summer and winter in a particular locality are 45°C and 15°C respectively. Find the value of Carnot COP for an air-conditioner

for cooling and heating, corresponding to refrigeration temperatures of 5°C for summer and heating temperature of 55°C for winter. Also find the theoretical power consumption per ton of refrigeration in each case.

Solution

For summer: $T_1 = 273 + 5 = 278$ K, $T_2 = 45 + 273 = 318$ K

For cooling, Carnot

Power consumption,

For winter: $T_1 = 273 + 15 = 288$ K, $T_2 = 273 + 55 = 328$ K

For heating,

Power consumption,

Example 18.5

A refrigerator has working temperature of –30°C and 35°C. The actual COP is 0.75 of the maximum. Calculate the power consumption, and heat rejected to the surroundings per ton of refrigeration.

Solution

Power consumption per ton

Heat rejected per ton = 3.5167 + 1.256 = 4.7727 kW

18.9 ❒ AIR REFRIGERATION CYCLES

There are two types of air refrigeration cycles.

1. Open air refrigeration cycle.
2. Closed (or dense) air refrigeration cycle.

18.9.1 Open Air Refrigeration Cycle

In an open air refrigeration cycle, the air is directly (at atmospheric pressure) led to the space to be cooled (i.e. refrigerator). The air is then allowed to circulate through the cooler and returned to the compressor to start another cycle. This requires size of the compressor and expander to be large. The moisture is regularly carried away by the air circulated through the refrigerator. This leads to the formation of frost at the end of expansion process and clog the line.

Therefore, a drier is required in the open cycle system.

18.9.2 Closed (or dense) Air Refrigeration Cycle

In a closed air refrigeration cycle, the air is passed through the pipes and components parts of the system at all times. The air is used for absorbing heat from the other fluid (say brine) which is circulated into the space to be cooled. The air does not come in contact directly with the space to be cooled.

This system has the following advantages over the open cycle:

1. The suction pressure is higher than that of atmospheric pressure Therefore, the volume of air handled by the compressor and expander is smaller.
2. The operating pressure ratio can be reduced, which results in higher coefficient of performance.

18.10 ❒ REVERSED CARNOT CYCLE

A reversed Carnot cycle, using air as working medium is shown in Fig. 18.5 on *p-v* and *T-s* diagrams. The reversed Carnot cycle is represented by four processes as described below:

1. **Process 1-2: Isentropic compression**

   During this process, the pressure of air increases from p_1 to p_2, specific volume decreases from v_1 to v_2 and temperature increases from T_1 to T_2. No heat is absorbed or rejected by air during this process.

2. **Process 2-3: Isothermal compression**

   During this process, the pressure of air increases from p_2 to p_3 and specific volume decreases from v_2 to v_3. Temperature remains constant at $T_2 = T_3$.

   Heat rejected by air per kg of air, $q_{2\text{-}3}$ = area 2-3-3′-2′

   $$= T_3 (s_2 - s_3) = T_2 (s_2 - s_3)$$

   Figure 18.5 *Reversed Carnot Cycle: (a) p-v diagram, (b) T-s diagram*

3. **Process 3-4: Isentropic expansion**

   During this process, the pressure of air decreases from p_3 to p_4, specific volume increases from v_3

to v_4 and temperature decreases from T_3 to T_4. No heat is absorbed or rejected by air.

4. **Process 4-1: Isothermal expansion**

   During this process, the pressure of air decreases from p_4 to p_1 and specific volume increases from v_4 to v_3. Temperature remains constant at $T_4 = T_1$.

Heat absorbed by air (or heat extracted from cold body) per kg of air.

$$q_{4-1} = \text{are } 4\text{-}1\text{-}2'\text{-}3'$$
$$= T_4\,(s_1 - s_4) = T_4\,(s_2 - s_3) = T_1\,(s_2 - s_3)$$

Work done during the cycle per kg of air, $w_{\text{cycle}} = q_{2-3} - q_{4-1}$ = are 1-2-3-4

$$= T_2\,(s_2 - s_3) - T_1\,(s_2 - s_3)$$
$$= (T_2 - T_1)\,(s_2 - s_3)$$

Coefficient of performance of refrigeration system,

18.10.1 Temperature Limitations for Reversed Carnot Cycle

For a reversed Carnot cycle.

where T_1 = lower temperature, and

T_2 = higher temperature.

The COP of the reversed Carnot cycle may be improved by the following methods:

1. Decreasing the higher temperature T_2 of hot body.
2. Increasing the lower temperature T_1 of cold body.

The temperature T_1 and T_2 cannot be varied at will, due to certain functional limitations. The lowest possible refrigeration temperature is $T_1 = 0$ (absolute zero) at which $(COP)_{ref} = 0$. The highest possible refrigeration temperature is $T_1 = T_2$ i.e. when the refrigeration temperature is equal to the

temperature of the surroundings at which, $(COP)_{ref} = \infty$. Thus reversed Carnot COP varies between 0 and ∞.

To obtain maximum possible COP in any application,

1. the cold body temperature T_1 should be as high as possible, and
2. the hot body temperature T_2 should be as low as possible.

The lower the refrigeration temperature required and higher the temperature of heat rejection to the surroundings, the larger is the power consumption of the refrigerating machine. Also, the lower is the refrigeration temperature required, the lower is the refrigerating capacity obtained.

The temperature T_2 in winter is less than T_2 in summer. Therefore, $(COP)_{ref}$ in

winter will be higher than $(COP)_{ref}$ in summer. In other words, the Carnot refrigerator works more efficiently in winter than in summer. Similarly, if the lower temperature fixed by the refrigeration application is high, the $(COP)_{ref}$ will be high. Thus a Carnot refrigerator used for making ice at 0°C (273 K) will have less $(COP)_{ref}$ than a Carnot refrigerator used for air-conditioning plant in summer at 20°C when the ambient air temperature is 40°C.

18.10.2 Vapour as a Refrigerant in Reversed Carnot Cycle

The reversed Carnot cycle can be made almost practical by operating in the liquid-vapour region of a pure substance as shown in Fig. 18.6 on *T-s* diagram.

1. **Process 1-2: Isentropic compression**

   The vapours during compression are wet and becomes dry saturated at the end of the process. Such a process is called *wet compression*. The temperature rises from T_1 to T_2.

   $$\text{Compression work per kg of refrigerant, } w_{1-2} = h_2 - h_1$$

2. **Process 2-3: Condensation**

   During this process the temperature remains constant at T_2. Heat is rejected from h_2 to h_3 and refrigerant gets converted to liquid at the end of process.

   $$q_{2-3} = h_2 - h_3 = (h_{fg})t_2$$

3. **Process 3-4: Isentropic expansion**

   During this process the flashing of the liquid refrigerant takes place with consequent temperature drop from T_2 to T_1. The refrigerant becomes wet at the end of expansion.

   Work of expander, $w_{3-4} = h_3 - h_4$

4. **Process 4-1: Evaporation**

   During this process the wet refrigerant evaporates, heat is absorbed from the medium, and refrigeration effect is produced.

Figure 18.6 *Reversed Carnot cycle with vapour as a refrigerant*

$$\text{Heat absorbed, } q_{4-1} = h_1 - h_4$$

Network, $w_{net} = w_{1-2} - w_{3-4} = (h_2 - h_1) - (h_3 - h_4)$

$$= (h_2 - h_3) - (h_1 - h_4) = q_{2-3} - q_{4-1}$$

Example 18.6

A Carnot refrigerator has working temperatures of –30°C and 35°C. It operates with R-12 refrigerant as a working substance. Calculate the work of isentropic compression, isentropic expansion, refrigeration effect, and COP of the cycle per kg of refrigerant.

If the actual COP is 80 percent of the maximum, calculate the power consumption and heat rejected to

the surroundings per ton of refrigeration.

Solution

Given: $t_1 = -30°C$, $t_2 = 35°C$, Refrigerant = *R*-12, $(COP)_{actual} = 0.8(COP)_{max}$ From the table of properties of *R*-12, we have (Refer to Fig. 18.6)

$$s_1 = s_2 = 0.6839 \text{ kJ/(kg.K)},\ s_3 = s_4 = 0.2559 \text{ kJ/(kg.K)},\ h_2 = 201.5 \text{ kJ/kg.}$$

$$h_3 = 69.5 \text{ kJ/kg},\ s_{f1} = s_{f4} = 0.0371 \text{kJ/(kg.K)},\ s_{g1} = s_{g4} = 0.7171 \text{ kJ/(kg.K)}$$

$$h_{f1} = h_{f4} = 8.9 \text{ kJ/kg},\ h_{g1} = h_{g4} = 174.2 \text{ kJ/kg}$$

Now $s_1 = s_{f1} + x_1 (s_{g1} - s_{f1}) = 0.0371 + x_1(0.7171 - 0.371)$

$$= 0.0371 + 0.68\, x_1 = 0.6839$$

$x_1 = 0.951$

$$h_1 = h_{f1} + x_1\,(h_{g1} - h_{f1}) = 8.9 + 0.951(174.2 - 8.9) = 166.1 \text{ kJ/kg}$$

$$s_4 = s_{f4} + x_4\,(s_{g4} - s_{f4}) = 0.0371 + x_4\,(0.7171 - 0.0371) = 0.2559$$

$x_4 = 0.3218$

$$h_4 = h_{f4} + x_4\,(h_{g4} - h_{f4}) = 8.9 + 0.3218\,(174.2 - 8.9) = 62.1 \text{ kJ/kg}$$

Work of compression, $w_{1-2} = h_2 - h_1 = 201.5 - 166.1 = 35.4$ kJ/kg

Work of expression, $w_{3-4} = h_3 - h_4 = 69.5 - 62.1 = 7.4$ kg

Refrigerating effect, $q_{4-1} = h_1 - h_4 = 166.1 - 62.1 = 104$ kJ/kg

Heat rejected, $q_{2-3} = h_2 - h_3 =$

201.5−69.5=132 kJ/kg

Network, $w_{net} = w_{1-2} - w_{3-4} = 35.4 - 7.4 = 28$ kJ/kg

Power consumption per ton of refrigeration

Heat rejected per ton of refrigeration to the surroundings = 3.5167 + 1.1761 = 4.6928 kW

18.10.3 Gas as a Refrigerant in Reversed Carnot Cycle

Figure 18.7 shows the *p-v* and *T-s* diagrams for the reversed Carnot cycle with a gas as a refrigerant.

1. **Process 1-2: Isentropic compression**

During this process, the pressure increases from p_1 to p_2 and volume decreases from v_1 to v_2. Heat transfer is zero so that $q_{1-2} = 0$. Temperature of gas increases from T_1 to $T_{2.}$

Work done per kg of gas,

2. **Process 2-3: Isothermal Compression**

   During this process, the pressure increases from p_2 to p_3 and volume decreases from v_2 to v_3. The temperature remains constant at T_2.

   Figure 18.7 *Reversed Carnot cycle with gas as a refrigerant: (a) p-v diagram, (b) T-s diagrame*

   Work done per kg of gas,

   Heat rejected, $q_{2-3} = w_{2-3}$ for a perfect gas

3. **Process 3-4: Isentropic expansion**

   Pressure decreases from p_3 to p_4, volume increases from v_3 to v_4 and temperature decreases from $T_2 = T_3$ to $T_4 = T_1$

4. **Process 4-1: Isothermal expansion**

   Pressure decreases from p_4 to p_1, volume increases from v_4 to v_1, and temperature remains constant at T_1.

Work done,

Refrigerating effect, $q_{4-1} = w_{4-1}$ for a perfect gas

Net work of the cycle, $w_{net} = w_{2-3} - w_{4-1}$

Refrigerating effect,

For the isentropic processes 1-2 and 3-4, we have

where r = compression ratio for the isentropic processes.

Thus, COP is a function of compression ratio only.

18.10.4 Limitations of Reversed Carnot Cycle

The limitations of reversed Carnot cycle are:

1. The isentropic compression and expansion processes require high speed while the isothermal condensation and evaporation processes require an extremely low speed. This variation in speed of air of a cycle is not practicable.
2. It is difficult to design an expander to handle a mixture of largely liquid and partly vapour.

3. Because of the internal irreversibilities in the compressor and the expander, the actual COP of the reversed Carnot cycle is very low.
4. With gas as refrigerant, it is not possible to devise, in practice, isothermal processes of heat absorption and rejection.
5. The stroke volume of gas cycle cylinder is very large resulting in poor actual COP.

18.11 ❐ BELL-COLEMAN CYCLE (OR REVERSED BRAYTON OR JOULE CYCLE)

The Bell-Coleman air refrigeration cycle was developed by Bell-Coleman and Light Foot by reversing the Joule's or Brayton's air cycle. The schematic diagram of such a cycle is shown in Fig. 18.8. It consists of a compressor, a cooler, an expander, and a refrigerator. The *p-v* and *T-s* diagrams are shown in Fig. 18.9.

1. **Process 1-2: Isentropic compression**

   The cold air from the refrigerator is drawn into the compressor cylinder where it is compressed isentropically. The pressure and temperature of air increases and the specific volume decreases from v_1 to v_2. No heat is absorbed or rejected by

the air.

2. **Process 2-3: Constant pressure cooling**

The warm air from the compressor is passed into the cooler where it is cooled at constant pressure $p_3 = p_2$. The temperature decreases from T_2 to T_3 and specific volume reduces from v_2 to v_3.

Heat rejected by air, $q_{2-3} = c_p(T_2 - T_3)$

3. **Process 3-4: Isentropic expansion**

The air from the cooler is drawn into the expander cylinder where it is expanded isentropically from p_3 to p_4. The temperature falls from T_3 to T_4 and specific volume increases from v_3 to v_4. No heat is absorbed or rejected by air during this process.

4. **Process 4-1: Constant pressure expansion**

The cold air from the expander is passed to the refrigerator where it is expanded at constant pressure $p_4 = p_1$. The temperature of air increases from T_4 to T_1 and specific volume increases from v_4 to v_1.

Heat absorbed by air (or extracted from the refrigerator), $q_{4-1} = c_p (T_1 - T_4)$

Figure 18.8 *Schematic diagram of Bell-Coleman air refrigeration cycle*

Figure 18.9 *Bell-Coleman cycle: (a) p-v diagram, (b) T-s diagrame*

Work done during the cycle per kg of air,

$$W = \text{Heat rejected} - \text{Heat absorbed}$$
$$= q_{2-3} - q_{4-1} = c_p[(T_2 - T_3) - (T_1 - T_4)]$$

For isentropic compression process and for isentropic expansion process 3-4,

Since $p_2 = p_3$ and $p_1 = p_4$, therefore,

where

18.11.1 Bell-Coleman Cycle with Polytropic Processes

Let the compression and expansion processes take place according to the polytropic law. $pv^n = \text{const.}$

1. **Process 1-2: Polytropic compression**

   Work done per kg of air,

2. 2. **Process 3-4: Polytropic expansion**

   Net work done during the cycle per kg of air,

3. Heat absorbed during constant pressure process 4-1.
   1. For $n = \gamma$, Eq. (18.17) reduces to Eq. (18.16*b*)
   2. For $n < \gamma$, $(COP)_{poly} > (COP)_{isen}$
   3. For $n > \gamma$, $(COP)_{poly} < (COP)_{isen}$

   Therefore, for obtaining higher COP, the compression and expansion index should be kept less than 1.4.

Example 18.7

A closed cycle refrigeration system working between 5 bar and 20 bar extracts 144 MJ of heat per hour. The air enters the compressor at 6°C and the expander at 22°C. The unit operates at 320 rpm. Calculate by assuming isentropic compression and expansion:

1. Power required to operate the unit.
2. Bore of compressor, and
3. Refrigerating capacity in tonnes of ice at 0°C per day.

The following data is available:

Solution

Given: $p_1 = p_4 = 5$ bar, $p_2 = p_3 = 20$ bar, $Q_{4\text{-}1} = 144$ MJ/h or 2400 kJ/min, $T_1 = 273 + 6 = 279$ K, $T_3 = 273 + 22 = 295$ K, $N = 320$ rpm, $L = 300$ mm, $\eta_c = 0.80$, $\eta_e = 0.85$, $= 1.4$

For *p*-*v* and *T*-*s* diagrams, refer to Fig. 18.9.

1. Heat extracted from the refrigeration system per kg of air,

$$q_{4-1} = c_p\,(T_1 - T_4) = 1.005\,(279 - 198.5) = 80.9 \text{ kJ/kg}$$

Mass of air circulated,

Work done during isentropic compression process 1-2,

Work done during isentropic expansion process 3-4,

Net work done per kg of air supplied to the system,

$$q_{net} = w_{1-2} - w_{3-4} = 170.26 - 82.39 = 87.87 \text{ kJ/kg}$$

Power required to operate the system,

2. $$p_1 V_1 = m_a R T_1$$

Also (for double acting compressor)

3. Refrigerating capacity of the system per day = $q_{4-1} \times m_a \times 60 \times 24$

$$= 80.9 \times 29.67 \times 60 \times 24 = 3456.436 \text{ MJ}$$

Assuming latent heat of ice = 336 kJ/kg

Ice formation capacity of system =

Example 18.8

A closed cycle reversed Brayton refrigerator is required for a capacity of 12*TR*. The cooler pressure is 4.5 bar and the refrigerator pressure is 1.5 bar. The air is cooled in the cooler at a temperature of 45°C and the temperature of air at inlet to compressor is 20°C. Calculate for the ideal cycle: (a) COP, (b) mass of air circulated per minute, (c) theoretical displacement of compressor piston, (d) theoretical displacement of expander piston, and (e) net power per tonne of refrigeration. The compression and expansion are isentropic.

Solution

Given: $Q = 12TR$, $p_1 = p_4 = 1.5$ bar, $p_2 = p_3 = 4.5$ bar, $T_1 = 273 - 20 = 253$ K, $T_3 = 273 + 45 = 318$ K.

The *p-v* and *T-s* diagrams are shown in Fig. 18.10.

1.

2. Heat extracted per minute, $Q = 211 \times 12 = 2532$ kJ/min

   Heat extracted from the refrigerator per kg of air,

$$q = c_p\,(T_1 - T_4) = 1.005(253 - 232.3) = 20.8 \text{ kJ/kg}$$

Figure 18.10 *Reversed Brayton cycle: (a) p-v diagram, (b) T-s diagram*

   Mass of air circulated,

3.
4.
5. Work done per minute = $\dot{m}_a$ (Heat rejected – Heat extracted)

$$= \dot{m}_a\, c_p\,[(T_2 - T_3) - (T_1 - T_4)]$$
$$= 121.73 \times 1.005\,[(346.3 - 318) - (253 - 232.3)]$$
$$= 929.77 \text{ kJ/min}$$

   Net power per tonne of refrigeration =

Example 18.9

Dense air is used as refrigerant in Reverse-Brayton or Bell-Coleman or Joule cycle. Draw *T-s* and *p-v* diagrams for the cycle. Derive the expression for COP in terms of pressure ratio. If temperatures at the end of heat absorption and heat rejection are 0°C and 30°C respectively, the pressure ratio is 4 and the pressure in the cooler is 4 bar determine the temperatures at all state points and volume flow rates at inlet to compressor and at exit of turbine for 1 TR cooling capacity.

Solution

The T-s and p-v diagrams are shown in Fig. 18.11(a) and the schematic diagram in Fig. 18.11(b).

$$q_r = q_{2-3} = c_p\,(T_2 - T_3)$$
$$q_a = q_{4-1} = c_{4-1}\,(T_1 - T_4)$$
$$\text{Work done} = q_r - q_a = c_p\,(T_2 - T_3) - c_p\,(T_1 - T_4)$$

Figure 18.11 *Bell-Coleman cycle: (a) p-v diagram, (b) T-s diagram, (c) Schematic diagram*

Heat extracted per minute = 1 × 211 = 211 kJ/min

Heat extracted from cold chamber per kg of air

$$= c_p(T_1 - T_4) = 1.005 \times (273 - 203.82) = 69.526 \text{ kJ/kg}$$

Mass flow rate of air,

Volume handled by the compressor,

Volume handled by expander,

Example 18.10

An air refrigeration used for food storage provides 30 TR. The temperature of air entering the compressor is 10°C and temperature at exit of cooler is 30°C. Calculate:

1. COP of the cycle, and
2. Power required per tonne of refrigeration by the compressor.

The quantity of air circulated in the system is 3600 kg/h. The compression and expansion processes follow the law $pv^{1.3}$ = const. and $\gamma = 1.4$, $c_p = 1.005$ kJ/kg.K for air.

Solution

Given: $Q = 30\ TR$, $T_1 = 273 + 10 = 283$ K, $T_3 = 273 + 30 = 303$ K, $\dot{m}_a = 3600$ kg/h, $n = 1.3$

1. Heat extracted from the refrigerator, $Q = 30 \times 211 = 6330$ kJ/min
2. Heat absorbed,

   Work done per minute,

   Power required per ton of refrigeration

Example 18.11

A dense air refrigeration cycle operates between pressures of 4 bar and 20 bar. The air temperature after heat rejection to surroundings is 40°C and air temperature at exit of refrigeration is 8°C. The

isentropic efficiencies of compressor and expander are 0.82 and 0.86 respectively. Determine the compressor and expander work per TR. COP, and power required per TR. Take $\gamma = 1.4$ and $c_p = 1.005$ kJ/kg.K.

Solution

Given: $p_1 = p_4 = 4$ bar, $p_2 = p_3 = 20$ bar, $T_3 = 273 + 40 = 313$ K, $T_1 = 273 + 8 = 281$ K, $\eta_c = 0.82$, $\eta_e = 0.86$, $\gamma = 1.4$, $c_p = 1.005$ kJ/kg.K.

The *p*-*v* and *T*-*s* diagrams are shown in Fig. 18.12

Process 1-2: Isentropic compression

Figure 18.12 *Dense air refrigeration cycle: (a) p-v diagram, (b) T-s diagram*

Process 3-4: isentropic expansion

Isentropic efficiency of compressor,

Isentropic efficiency of expander,

Refrigerating effect per kg of air, $q_{4'-1} = c_p(T_1 - T_{4'}) = 1.005\ (281 - 213.7) = 67.6$ kJ/kg

Mass flow rate of air,

Compressor work per TR, $W_c = \dot{m}_a c_p\ (T_{2'} - T_1) = 3.12\times\ 7.005(481-281)= 627.12$ kJ/min

Expander work per TR, $W_e = \dot{m}_a c_p (T_3 - T_{4'}) = 3.12 \times 1.005(313-213.7) = 311.36$ kJ/min

Net work done, $W_{net} = W_c - W_e = 627.12 - 311.36 = 315.76$ kJ/min

Power required per TR =

18.12 ❒ REFRIGERANTS

A refrigerant may be defined as a substance which absorbs heat through expansion or vaporization and loses it through condensation in a refrigeration system.

18.13 ❒ CLASSIFICATION OF REFRIGERANTS

1. **Primary refrigerants:** They take part directly in refrigeration and cool the substance on absorption of their latent heat. Some of the primary refrigerants are NH_3, CO_2CH_3, $C1_2$, CH_2C1_2, SO_2, C_2H_5C1 and Freon group.
2. **Secondary refrigerants:** They are first cooled with the help of primary refrigerant and then used for cooling other substances by absorption of their sensible heat only e.g. ice, solid CO_2.

Primary refrigerants may be grouped as follows:

1. **Halo-Carbon compounds:** They contain one or more of three halogens, fluorine, chlorine and bromine. Commercially they are sold under the names as Freon, Genetron, Irontron and Areton, They are used in domestic commercial and industrial refrigerating devices. Some of their important examples are:

2. **Azeotropes:** They are the mixtures of different refrigerants. They do not under go any separation with changes in temperature and pressure, e.g.

   R-500 (73.8% R12 + 26.2% R-152) :
   $CC1_2F_2 + CH_3CH\ F_2$

   R-502 (48.8% R22 + 51.2% R-115) :
   $CHC1F_2 + CC1\ F_2\ CF_3$

3. **Hydrocarbons:** They are hydrocarbons, e.g.

4. **Inorganic compounds:**

5. **Unsaturated organic compounds:** They contain

ethylene or propylene as main constituents, e.g.

18.14 ❒ DESIGNATION OF REFRIGERANTS

1. For a compound derived from a saturated hydrocarbon denoted by the chemical formula

$$C_mH_nF_pC1_q$$

   In which $(n + p + q) = 2m + 2$, the complete designation of a refrigerant is

$$R(m - 1)(n + 1)(p)$$

2. The brominated refrigerants are denoted by putting an additional B and a number to denote as to how many chlorine atoms are replaced by bromine atoms.
3. In the case of isomers, i.e., compounds with the same chemical formula but different molecular structure, subscrips a, b, etc. are used after the designations.
4. Unsaturated compounds, for which $(n + p + q) = 2m$, are distinguished by putting the digit 1 before $(m - 1)$.
5. For inorganic refrigerants, numerical designations have been given according to then molecular weight added to 700.
6. In the case of butane, C_4H_{10} and higher hydrocarbons, arbitrary designation R 600 is used.

18.15 ❒ DESIRABLE PROPERTIES OF REFRIGERANTS

A good refrigerant should have the following properties:

1. *Thermodynamic properties*
    1. Low boiling point
    2. Low freezing point

3. High saturation temperature
4. High latent heat of vaporization
5. High critical pressure and temperature.

2. *Chemical properties*
1. Non-toxic
2. Non-flammable
3. Non-explosive
4. Non-corrosive
5. Chemically stable
6. Odourless

3. *Physical properties*
1. Low specific volume
2. High thermal conductivity
3. High electrical insulation
4. Low specific heat
5. Miscibility with lubricating oil.

4. *Other properties*
1. Ease of leakage detection
2. Ease of handling
3. High COP
4. Ease of availability and of low cost.
5. Low power consumption per ton of refrigeration.

18.16 ❒ APPLICATIONS OF REFRIGERANTS

The refrigerants and their applications are given in Table 18.1.

The primary-refrigerants are further classified into the following four groups

1. Halo-carbon refrigerants
2. A zoetrope refrigerants.
3. Inorganic refrigerants, and
4. Hydro-carbon refrigerants.

1. **Halo-carbon Refrigerants:** The commonly used halo-carbon refrigerants, their applications and method of leak testing are given in Table 18.2.
2. **Azeotrope Refrigerants:** The term 'azeotrope' refers to a stable mixture of refrigerants whose vapour and liquid phases retain identical compositions over a wide range of temperatures.

   Some of the azeotropic refrigerants, their applications and method of leak testing are given in Table 18.3.

Table 18.1 *Refrigerants and their applications*

Table 18.2 *Commonly used halo carbon refrigerants*

Table 18.3 *Azeotrope refrigerantss*

1. 3. **Inorganic Refrigerants:** The various inorganic refrigerants, their applications and leak testing method are given in Table 18.4.
2. 4. **Hydro-Carbon Refrigerants:** The various hydro-carbons are given in Table 18.5. They are mostly used for industrial and commercial installations.

Table 18.4 *Inorganic refrigerants*

Table 18.5 *Hydrocarbon refrigerants*

Table 18.6 *Ozone Depletion Potential (ODP) of Refrigerants*

1. 5. **Ozone Layer Depletion:** Stratosphere is about 25-30 km away from the earth's surface. It contains ozone (O_3). Ozone is formed in the stratosphere by the reaction that occurs when sunlight interacts with oxygen. This ozone shelters the earth from ultraviolet (UV)

radiation. This protection is essential for human health. This ozone is easily destroyed by the UV radiation. The sun's radiation will break down the O_3 molecule into a standard O_2 molecule and elemental free oxygen (O) atom. More O_3 is produced through photosynthesis and the bonding of O_2 with free oxygen O. Thus ozone is constantly being formed and destroyed in the stratosphere.

In the stratosphere, UV radiation will break off a chlorine atom (Cl) from the CFC and HCFC molecule. The chlorine atom destroys many O_3 molecules The depleted O_3 molecules in the stratosphere let more UV radiation reach the earth This causes an increase in skin cancers and frequency of cataracts in humans; weakens human immune system and decreases plant and marine life. Ozone depletion potential (ODP) has been used as an index to show the effect of a refrigerant on O_3. Lower the ODP, it will be better. The ODP of some of the refrigerants is given in Table 18.6.

18.17 ❐ ECO-FRIENDLY REFRIGERANTS

With the increasing awareness of environmental degradation, the production, use and disposal of Chloro Fluoro Carbons (CFCs) and Hydro Chloro Fluoro Carbons (HCFCs) as refrigerants in mechanical refrigeration

systems has become a subject of great concern. However, such systems are being developed using more eco-friendly refrigerants, such as

1. Hydrocarbons (HG-290, HC-600a).
2. Ammonia (NH_3)
3. Carbon Dioxide (CO_2)
4. Water (R-718)
5. Air.

Eco-friendly refrigerants are also known as natural refrigerants. The important features and applications of eco-friendlv refrigerants are given in Table 18.7.

Table 18.7 *Important features and applications of eco-friendly refrigerants*

18.18 ❒ REFRIGERANT SELECTION

The choice of a refrigerant depends upon the following factors:

1. Refrigeration temperature required.
2. Refrigerating capacity of plant.
3. Thermodynamic requirements: Normal boiling point, condensing and evaporating pressure, critical temperature and pressure, freezing point, volume of suction vapour per ton, COP, power consumption per ton etc
4. Chemical requirements: Flammability, toxicity, action with water and oil (mixibility), and action with materials of construction
5. Physical requirements: Dielectric strength, thermal conductivity, viscosity, heat capacity, surface tension, leak tendency and cost of refrigerant
6. Economies.
7. Equipment type and size.
8. Application.

Multiple-choice Questions

1. One tonne of refrigeration is equal to
    1. 211 kJ/min
    2. 220 kJ/min
    3. 420 kJ/min
    4. 620 kJ/min
2. One tonne refrigerating machine means that
    1. one tonne is the total mass of the machine
    2. one tonne of refrigerant is used
    3. one tonne of water can be converted into ice
    4. one tonne of ice when melts from and at 0°C m 24 hours, the refrigeration effect produced is equivalent to 211 kJ/min.
3. The ratio of heat extracted in the refrigerator to the work done on the refrigerant is called
    1. coefficient of performance of refrigeration
    2. coefficient of performance of heat pump
    3. relative coefficient of performance
    4. refrigerating efficiency
4. The relative coefficient of performance is equal to
    1.
    2.
    3. Actual COP × Theoretical COP
5. Air refrigerator works on
    1. Carnot cycle

   2. Rankine cycle
   3. reversed Carnot cycle
   4. Bell-Coleman cycle
6. In air-conditioning of aeroplanes, using air as a refrigerant. the cycle used is
   1. reversed Carnot cycle
   2. reversed -Joule cycle
   3. reversed Brayton cycle
   4. reversed Otto cycle
7. Co-efficient of performance of a Reversed Carnot cycle refrigerator working between higher temperature T_2 and lower temperature T_1
   1. will increase with increase in T_1 keeping T_2 fixed
   2. will decrease with increase in T_1 keeping T_2 fixed
   3. will first increase with increase in T_1 and then decrease with increase in T_1 keeping T_2 fixed
   4. None of the above.
8. A reversible refrigerator working between two fixed temperatures
   1. has the same COP whatever be the working substance
   2. has its COP increased for working substance with high enthalpy of evaporation
   3. has its COP increased for working substance with higher specific heats
   4. none of the above.
9. Reversed Carnot cycle comprises
   1. two isentropic processes and two adiabatie processes
   2. two isentropic processes and two isothermal processes
   3. two isentropic processes and two constant pressure processes
   4. two isentropic processes and two constant volume processes.
10. In Reversed Carnot cycle working on perfect gas
   1. Isothermal work of compression is equal to isothermal work of expansion
   2. Isentropic work of compression is equal to isentropic work of expansion
   3. Net work of the cycle is zero
   4. Net heat transfer of the cycle is zero.
11. In a Reversed Carnot-cycle working on vapour
   1. Isentropic work of compression is equal to isentropic work of expansion

2. Isothermal heat absorption is equal to isothermal heat rejection
3. There is no work done during isothermal processes
4. There is no work done during isentropic processes

12. Reversed Carnot cycle assumes that all processes in the cycle are
    1. Non-flow only
    2. Steady flow only
    3. Non-flow or steady flow
    4. Transient flow
13. Two Carnot refrigerators are employed, one for ice making and other for comfort cooling
    1. The COP of refrigerator for ice making is higher than that for the other
    2. The COP of refrigerator for ice making is lower than that for the other
    3. The COP of refrigerator for ice making is same as that for the other
    4. The COP of Carnot refrigerator will depend on refrigerant used
14. A reverse Carnot cycle has a COP of 4. The ratio of higher temperature to lower temperature will be
    1. 1.5
    2. 2
    3. 1.25
    4. 2.5
15. A Carnot refrigerator requires 70 kJ/min of work to produce one ton of refrigeration at -40°C. The COP of this refrigerator is
    1. 4
    2. 3
    3. 5
    4. not possible to find
16. The coefficient of performance of refrigerator working on Reversed Carnot cycle with T_1 being lower temperature and T_2 being higher temperature is
    1.
    2.
    3.
    4.
17. The coefficient of performance of Carnot refrigerator can be expressed (r = Volume compression ratio for isentropic compression)
    1.

2.
3.
4.

18. The COP of Carnot Refrigerator is 3 and it produces 1 ton of refrigeration. The work that will be done is equal to
    1. 210 kJ/min
    2. 70 kJ/min
    3. 100 kJ/min
    4. 200 kJ/min.
19. The ratio of high temperature to low temperature for Reversed Carnot refrigerator is 1.25. The COP will be
    1. 2
    2. 3
    3. 4
    4. 5
20. A carnot refrigerator operates between 300.3K and 273K. The fraction of cooling effect required as work input is
    1. (a) 20%
    2. (b) 10%
    3. (c) 50%
    4. (d) not possible to find with the data
21. The relationship between $COP_{heat\ pump}$ $COP_{refrigerator}$ for the same range of temperature operation is
    1. $COP_{heat\ pump} = COP_{refrigerator}$
    2. $COP_{heat\ pump} - COP_{refrigerator} = 1$
    3. $COP_{heat\ pump} = 2COP_{refrigerator}$
    4. not possible to predict without knowing working substance.
22. Carnot refrigerator absorbs heat at −13°C and requires 1 kW for each 6.5 kW of heat absorbed, the COP and temperature of heat rejections respectively are (0°C = 273K)
    1. COP = 6.5, t = 27°C
    2. COP = 7.5. t = 27°C
    3. COP = 6.5, t = 30°C
    4. COP = 7.5. t = 37°C

Review Questions

1. How is the eff ectiveness of a refrigeration system measured?
2. Explain the term “tonne of refrigeration”.
3. Discuss the advantages of the dense air refrigerating system over an open air refrigeration system.

4. What is the diff erence between a refrigerator and heat pump? Derive an expression for coeffi cient of performance for both if they are running on reversed Carnot cycle.
5. Discuss the main differences between Reversed Carnot cycle operating on perfect gas and wet vapour.
6. What is refrigeration?
7. List few applications of refrigeration.
8. Defi ne coeffi cient of performance.
9. What are the SI units of refrigeration?
10. Enumerate the various methods of refrigeration.
11. Name few refrigeration systems.
12. What is refrigeration effect?

Exercises

18.1 A Reversed Carnot Cycle operates on the following substances given below and the range of temperatures is 300K and 250K.

1. Working substance is water between the range of dry saturated liquid to dry saturated vapour at 300K.
2. Working substance is water in the form of super heated steam with the range of pressure from 0.01 bar to 0.05 bar.
3. Working substance is CO_2 in the super heated region with pressure range of 0.5 bar to 5 bar
4. Working substance is air with pressure range of 1 bar to 10 bar

For the same heat rejected from the system of 60000 kJ/hr, determine the work input. Carnot coefficient of performance as refrigerator and as heat pump and the mass flow rate.

18.2 Carnot heat engine draws heat from a reservoir at temperature of 600 K and rejects heat to another reservoir at temperature *T*. This engine drives a Carnot reverse cycle refrigerator which absorbs heat from reservoir at temperature 200 K and reject heat to reservoir at temperature T. Determine the temperature *T* such that heat supplied to Carnot heat engine equal to heat absorbed by Carnot refrigerator.

18.3 A refrigerator using Carnot cycle requires 1.25 kW per tonne of refrigeration to maintain a temperature of –30°C. Find: (*a*) COP of the Carnot refrigerator, (*b*) temperature at which heat is rejected; and (*c*) Heat rejected per tonne of refrigeration.

[**Ans.** 2.8; 55.4°C; 284. kJ/min]

18.4 A Carnot cycle machine operates between the temperature limits of 47°C and 30°C. Determine the COP when it operates as (*a*) a refrigerator (*b*) a heat pump

[**Ans.** 3.16; 4.16]

18.5 Ten tonnes of fish is frozen to –30°C per day. The fish enters the freezing chamber at 30°C and freezing occurs at –3°C. The frozen fish is cooled to – 30°C. The specific heats of fresh and frozen fish are 3.77 kJ/kg K and 1.67 kJ/kg K respectively while latent heat of freezing is 251.2 kJ/kg K. Find the tonnage of the plant which runs for 18 hours per day. The evaporator and condensor temperatures are –40°C

and 45°C respectively. It the COP of the plant is 1.8, determine the power consumption of the plant in kW.

[**Ans.** 18.6 TR; 36.1 kW]

18.6 A Carnot refrigeration system has working temperature of –30°C and 40°C. what is the maximum COP possible? If the actual COP is 75% of the maximum calculate the actual refrigerating effect produced per kilowatt hour.

[**Ans.** 3.47; 0 713 TR]

18.7 A 5 tonne refrigerating machine operating on Bell Coleman cycle has an upper limit of pressure of 12 bar. The pressure and temperature at the start of compressor are 1 bar and 17°C respectively. The compressed air cooled

at constant pressure to a temperature of 40°C enters the expansion cylinder. Assuming both the expansion and compression processes to be isentropic with $\gamma = 1.4$; Determine *(a)* COP; *(b)* the quantity of air in circulation per minute; *(c)* piston displacement of compressor and expander *(d)* bore of compressor and expansion cylinders. The unit runs at 250 r.p.m. and is double acting. Stroke length is 200 mm; and *(e)* power required to drive the unit. Take $c_p = 1$ kJ/kg K; $c_v = 0.71$ kJ/kg K; R = 0.287 kJ/kg K.

[**Ans.** 0.952; 7.65 kg/min; 6.37 m^3/min; 3.35 m^3 /min; 284 mm; 18.4 kW]

18.8 An air refrigerator used for food storage provides 50 TR. The temperature of air entering the

compressor is 7°C and the temperature before entering into the expander is 27°C. Assuming a 70% mechanical efficiency find (*a*) actual COP and (*b*) The power required to run the compressor.

The quantity of air circulated in the system is 100 kg/min. The compression and expansion follow the law $pv^{1.3}$ = constant. Take $\gamma = 1.4 : c_p = 1$ kJ/kg K for air.

[**Ans.** 1.13; 110.6 kW]

18.9 A dense air refrigerating system operating between pressures of 17.5 bar and 3.5 bar is to produce 10 tonnes of refrigeration. Air leaves the refrigerating coils at –7°C and it leaves the air cooler at 15.5°C. Neglecting losses and

clearance, calculate the net work done per minute and the coefficient of performance. For air $c_p = 1.005$ kJ/kg K and $\gamma = 1.4$

[**Ans.** 1.237 kJ/min; 1.7]

18.10 A refrigerating plant is required to produce 2.5 tonnes of ice per day at –4°C from water at 20°C. If the temperature range in the compressor is between 25°C and –6°C, calculate power required to drive the compressor. Latent heat of ice = 335 kJ/kg and specific heat of ice = 2.1 kJ/kg K.

[**Ans.** 1 437 kW]

18.11 A refrigerator storage is supplied with 30 tonnes of fish at a temperature of 27°C. The fish has to be cooled to –9°C for preserving it for long period

without determination. The cooling takes place in 10 hours. The specific heat of fish is 2.93 kJ/kg K above freezing point of fish and 1.26 kJ/kg K below freezing point of fish which is –3°C. The latent heat of freezing is 232 kJ/kg. What is the capacity of the plant in tonnes of refrigeration for cooling the fish? What would be the ideal COP between this temperature range? If the actual COP is 40% of the ideal, find the power required to run the cooling plant.

[**Ans.** 78 TR; 7.33; 93.3 kW]

18.12 An air refrigeration plant working on the Carnot cycle between the temperature limits of 34°C and –1°C has a capacity of 10 ton. Determine the power input to the plant.

18.13 An air refrigeration plant working on Carnot cycle produces refrigeration capacity equivalent to production of 10 tons of ice in 24 hours at 0°C from water at 20°C. The compressor temperature limits are 30°C and –15°C. Determine the power input to the plant. The latent heat of ice is 335 kJ/kg.

18.14 Find the minimum power input to perfect reversed heat engine that will produce 1 ton of ice per hour at –5°C from water at 15°C. The temperature limits of the reversed heat engine are 15°C and –5°C. Assume specific heat of ice as 2.1 kJ/kg. K and latent heat of ice as 335 kJ/kg.

ANSWERS TO MULTIPLE-CHOICE QUESTIONS

1. a
2. d

3. a
4. b
5. d
6. c
7. a
8. a
9. b
10. b
11. c
12. c
13. b
14. c
15. b
16. b
17. c
18. c
19. c
20. b
21. b
22. a

Chapter 19

Vapour Compression and Vapour Absorption Systems

19.1 ❒ INTRODUCTION

The vapour compression refrigeration system is based upon the fact that fluids absorb heat while changing from a liquid phase to vapour phase and reject heat while changing from a vapour phase to a liquid phase. The temperature during the phase change remains constant. However, the temperature varies with the pressure and the fluid. In vapour compression system, the refrigerant condenses and evaporates at temperatures and pressures close to the atmospheric conditions. The closed

refrigeration cycle is used in which refrigerant does not leave the system. The refrigerant alternately condenses and evaporates. During evaporating the refrigerant absorbs its latent heat from the brine (salt water), which is used for circulating it around the cold chamber. While condensing, it gives out its latent heat to the circulating water of the cooler. The vapour compression system is used in a small domestic refrigerator to big industrial refrigeration systems.

19.2 ❐ COMPARISON OF VAPOUR COMPRESSION SYSTEM WITH AIR REFRIGERATION SYSTEM

Following are the advantages and disadvantages of the vapour compression refrigeration system over air refrigeration system:

Advantages

1. It has smaller size for the given capacity of refrigeration.
2. It has less running cost.
3. It can be employed over a large range of temperatures.
4. The coefficient of performance is quite high.

Disadvantages

1. The initial cost is high.
2. The prevention of leakage of the refrigerant is the major problem in vapour compression system.

19.3 ❒ SIMPLE VAPOUR COMPRESSION REFRIGERATION SYSTEM

The schematic diagram of a simple vapour compression refrigeration system is shown in Fig. 19.1. It consists of the following parts:

1. Compressor.
2. Condenser
3. Throttle valve
4. Evaporators.

1. **Compressor:** The low pressure and temperature vapour refrigerant from the evaporator is drawn into the compressor through the suction valve. It is compressed to a high pressure and temperatures in the compressor. The high pressure and temperature vapours are discharged into the condenser through

the delivery valve.

2. **Condenser:** The condenser or cooler consists of coils of pipes in which the high pressure and temperature vapour refrigerant is cooled and condensed. The refrigerant, while passing through the condenser coils gives up its latent heat to the condensing medium. The condensing medium is normally air or water.
3. **Throttle Valve:** Its function is to allow the liquid refrigerant under high pressure and temperature to pass at a controlled rate after reducing its pressure and temperature. Some of the liquid refrigerant evaporates. The enthalpy of the refrigerant remains unaltered in passing through the throttle valve.
4. **Evaporator:** It consists of coils of pipe in which the liquid-vapour refrigerant at low pressure and temperature is evaporated and converted into vapour refrigerant at low pressure and temperature. During evaporation, the liquid-vapour refrigerant absorbs its latent heat of vaporisation from the medium (air, water or brine), which is to be cooled.

Figure 19.1 *Schematic diagram of simple vapour compression refrigeration system*

19.4 ❒ VAPOUR COMPRESSION REFRIGERATION SYSTEM

A schematic diagram of a vapour compression refrigeration system is shown in Fig. 19.2. It consists of a compressor, condenser, expansion device for throttling and an evaporator. In plants with a large amount of refrigerant charge, a receiver and a drier is installed in the liquid line. The *p*-*v*, *T*-

s, and *p-h* diagrams for the system are shown in Fig. 19.3. The sequence of operations is as given below:

1. The process 1-2 is isentropic compression.

   $s_2 = s_1$, $Q = 0$. Work done by the compressor per kg of refrigerant

   $$W_{1\text{-}2} = -\int v\,dp = -\int dh = -(h_2 - h_1)$$

2. The process 2-3 is constant pressure condensation.

   Heat rejected, $Q_r = h_2 - h_3$

3. The process 3-4 is isenthalpic expansion.

   $$h_3 = h_4 = h_{f4} + x(h_1 - h_{f4})$$

   or Dryness fraction,

4. The process 4-1 is constant pressure evaporation. Refrigeration effect or Heat absorbed,

   $$Q_a = h_1 - h_4$$

Figure 19.2 *Vapour compression system*

Figure 19.3 *Vapour compression cycle: (a) p-v diagram, (b) T-s diagram, (c) p-h diagram*

1. From Fig. 19.3(b), we have for vapours dry and saturated at the beginning of compression and superheated at the end of compression.

Refrigerating effect, Q_a = area 1–4–6–5–1

Heat rejected, Q_r = area 2–2′–3–7–5–2

Work done, $W = Q_r - Q_a$

= area 1–2–2′–3–7–6–4–1

Area 4–4′–7–6–4 = area 3–0–4′–3

2. When vapours are dry and saturated at the end of compression, as shown in Fig. 19.4.

Work done = area 1–2–3–0–1 = Heat at 2 − Heat at 1 = $h_2 - h_1$

Refrigeration effect = area 1–4–6–5 = Heat at 1 − Heat at 4 = $h_1 - h_4$

Area 3–0–4′–3 = area 4–4′–7–6–4

∴ Work done, W = area 1–2–3–7–6–4–1

Dryness fraction at suction,

Figure 19.4 *Vapour compression cycle: (a) T-s diagram, (b) p-h diagram*

The COP of vapour compression cycle is lower than that of reversed Carnot cycle. Power per ton of refrigeration,

where v_1 = specific volume of the refrigerant at the inlet of compressor

D = dia of compressor piston or cylinder

L = stroke length

N = rpm

n = number of cylinders in compressor

η_v = volumetric efficiency

c = clearance ratio

Mass flow rate of cooling medium in the condenser:

19.5 ❐ USE OF *T-s* AND *p-h* CHARTS

The *T-s* (Temperature-entropy) diagram for the vapour compression cycle is shown in Fig. 19.5.

Figure 19.5 *T-s diagram for vapour compression cycle*

Figure 19.6 *p-h diagram for vapour compression cycle*

Figure 19.7 *Pressure-enthalpy (p-h) chart*

Process 1-2: Isentropic compression. State 1 represents saturated vapour and state 2 is super heated vapour.

Process 2-3: Condensation. State 2 represents saturated vapour and state 3 is saturated liquid. Heat is rejected to the condensing medium.

Process 3-4: Throttling process. The enthalpy of saturated liquid at state 3 is equal to the enthalpy at state 4 or 1.

Process 4-1: Evaporation. Heat is absorbed from the medium and refrigeration effect is produced.

The *p-h* (pressure-enthalpy) diagram is shown in Fig. 19.6. The various

processes are also shown in this figure. The *p-h* chart is shown in Fig. 19.7. It is the most convenient chart for studying the behaviour of a refrigerant. The vertical ordinates represent pressure and horizontal ordinates represent enthalpy. The saturated liquid line and the saturated vapour line merge into one another at the critical point. The space to the left of the saturated liquid line is the sub-cooled liquid region. The space between the liquid and vapour lines is called the wet region. The space to the right of the saturated vapour line is the superheated vapour region.

Example 19.1

An *R*-12 vapour compression

system is operating at a condenser temperature of 40°C and an evaporator temperature of –5°C. Calculate the Carnot COP and actual COP of the cycle.

Solution

From *p-h* diagram of *R*-12, we have

$$h_1 = 185.4 \text{ kJ/kg},\ h_2 = 208.0 \text{ kJ/kg},\ h_3 = h_4 = 74.6 \text{ kJ/kg}$$

Carnot

Actual

Example 19.2

An ammonia refrigerating machine has working temperature of 35°C in

the condenser and −15°C in the evaporator. Calculate the COP for (a) dry Compression, and (b) wet compression. The data for ammonia is given below:

Solution

1. By interpolation for entropy,

   Degree of superheat at discharge = 50 + 27.1 = 77.1°C

   Discharge temperature = 35 + 77.1 − 112.1°C

   Refrigerating effect, $Q_a = h_1 - h_4 = 1426 - 347.5 = 1078.5$kJ/kg

   Work done, $W = h_2 - h_1 = 1685.4 - 1426 = 259.4$ kJ/kg

2. $$h_1 = 112.3 + 0.88(1426 - 112.3) = 1268.4 \text{ kJ/kg}$$

$$v_1 = v_{f1} + x_1\,(v_{g1} - v_{f1})$$

$$= 0.00152 + 0.88(0.509 - 0.00152) = 0.448 \text{m}^3\text{/kg}$$

$$Q_a = h_1 - h_4 = 1268.4 - 347.5 = 920.9 \text{ kJ/kg}$$

$$W = h_2 - h_1 = 1471 - 1268.4 = 202.6 \text{ kJ/kg}$$

Example 19.3

A refrigerating plant using CO_2 as refrigerant works between 25°C and –5°C. The dryness fraction of CO_2 is 0.6 at the entry of compressor. Find the ice formed per day if the relative efficiency is 50%. Ice is formed at 0°C from water at 10° C. The quantity of CO_2 circulated is 6 kg/min. For water c_p = 4.187 kJ/kg, L = 335 kJ/kg.

Properties of CO_2

Solution

The T-s cycle is shown in Fig. 19.8.

$$T_1 = 273 - 5 = 268\text{K},$$
$$T_2 = 273 + 25 = 298\text{K}$$
$$h_1 = h_{f1} + x_1 h_{fg1} = -7.53 + 0.6\ 245.8$$
$$= 140 \text{ kJ/kg}$$
$$s_1 = s_2$$
$$x_2 = 0.63$$
$$h_2 = h_{f2} + x_2 h_{fg2} = 81.25 + 0.63 \times 121.5 = 157\text{kJ/kg}$$

Work done, $w_{1-2} = h_2 - h_1 = 157 - 140 = 17\text{kJ/kg}$

Figure 19.8 *T-s diagram*

Refrigerating effect, $q_a = h_1 - h_4 = h_1 - h_3 = h_1 - h_{f3}$

Actual COP = 3.45 × 0.5 = 1.725

Work done per second =

Actual cooling effect = 1.725 × 1.7 = 2.94 kJ/s

Heat carried to form 1 kg of ice

$= c_p\,(t_2 - t_1) + L = 4.187 \times 10 + 335 = 376.87$ kJ

Ice formed per hour =

Example 19.4

A refrigerator operating on a standard vapour compression cycle has a coefficient of performance of 6.5 and is driven by a 50 kW compressor. The enthalpies of saturated liquid and saturated vapour refrigerant at the operating condensing temperature of 35°C are 69.55 kJ/kg and 201.45kJ/kg respectively. The saturated refrigerant vapour leaving the evaporator has an enthalpy of 187.53kJ/kg. Find the refrigerant temperature at the compressor

discharge. The c_p of refrigerant vapour may be taken to be 0.6155kJ/kg.K.

Solution

COP = 6.5, Compressor power = 50 kW

Refrigerating capacity = 50 × 6.5 = 325 kW

Heat extracted per kg of refrigerant = 187.53 – 69.55 = 117.98 kJ/kg

Enthalpy of vapour after compression = 187.53 + 18.15 = 205.68 kJ/kg

Superheat = 205.68 – 201.45 = 4.23

kJ/kg

Temperature of refrigerant at compressor discharge = 35° + 6.87° = 41.87°C

Example 19.5

A refrigeration system of 15 tons capacity operates on standard simple vapour compression cycle using Refrigerant -22 to an evaporator temperature of 5°C and condensing temperature 50°C. Draw the *p-h* diagram for the cycle. Calculate (a) the refrigerant mass flow rate, and (b)the compressor intake volume flow rate if the

compressor volumetric efficiency is 0.72. Use the refrigerant property data given in Table below.

Figure 19.9 *(a) Simple vapour compression system, (b) p-h diagram*

Solution

Refer to Fig. 19.9.

1. Capacity of plant

$$h_1 = 407.1 \text{ kJ/kg},$$
$$h_4 = h_3 - 263.3\text{kJ/kg}$$

2. Volume flow rate

Example 19.6

A refrigeration cycle uses Freon –12 as the working fluid. The temperature of the refrigerant in the evaporator is –10°C. The condensing temperature is 40°C.

The cooling load is 150 W and the volumetric efficiency of the compressor is 80%. The speed of compressor is 720 rpm. Calculate the mass flow rate of the refrigerant and the displacement volume of the compressor.

Properties of Freon – 12

Solution

Cooling load Q = 150 W

$$\begin{aligned} \text{Refrigeration effect} &= h_1 - h_4 \text{ (Fig.19.10)} \\ &= 183 - 74.5 \\ &= 108.5\text{kj/kg} \\ \dot{m}(h_1 - h_4) &= Q \end{aligned}$$

Figure 19.10 *p-h diagram*

Mass flow rate of refrigerant,

Displacement volume of compressor,

Example 19.7

In a simple vapour compression cycle, following are the properties of the refrigerant R-12 at various points.

The piston displacement volume for the compressor is 1.5 litres per stroke and its volumetric efficiency is 80%. The speed of the compressor is 1600 rpm.

Find (a) the power rating of the compressor (kW), (b) the refrigerating effect (kW).

Solution

Piston displacement volume (Fig. 19.11),

Figure 19.11 *p-h diagram*

Compressor discharge,

Mass flow rate of compressor,

$= 39.12 \text{kg/min}$

Power of compressor = $\dot{m}(h_2 - h_1)$

Refrigerating effect = $\dot{m}(h_1 - h_4)$ = 39.12(183.2 – 84.9) = 3845.5 kJ/

min or 64.1kW

Example 19.8

In a standard vapour compression refrigeration cycle operating between an evaporator temperature of –10°C and condenser temperature of 40°C, the enthalpy of the refrigerant, Freon-12, at the end of compression is 220 kJ/kg. Show the cycle diagram on *T*-*s* plane.

Calculate (a) the COP of the cycle, (b) the refrigerating capacity and the compressor power assuming a refrigerant flow rate of 1 kg/min. You may use the extract of the

Freon-12 property table given below:

Solution

The *T-s* diagram is shown in Fig. 19.12

Refrigerating capacity

$$= \dot{m}\,(h_1 - h_4)$$
$$= 1(183.1 - 26.85)$$
$$= 156.25 \text{ kJ/min or } 2.6 \text{ kW}$$
$$\text{Compressor power} = \dot{m}\,(h_2 - h_1)$$
$$= 1\,(220 - 183.1)$$
$$= 36.9 \text{ kJ/min or } 0.615 \text{ kW}$$

Figure 19.12 *T-s diagram*

Example 19.9

An R-717 (ammonia) system

operates on the basic vapour compression refrigeration cycle. The evaporator and condenser pressures are 0.119 MPa and 1.389 MPa respectively. The mass flow rate of refrigerant is 0.1 kg/s. The volumetric efficiency of the compressor is 84%. Determine the compressor displacement rate. If the COP of the cycle is 2, determine the power input to the compressor.

Saturation Properties of R-717 (ammonia)

Solution

Compressor displacement (Fig. 19.13)

Refrigerating effect = $h_1 - h_4$

$$= 1423.6 - 371.4$$
$$= 1052.2 \text{ kJ/kg}$$

Figure 19.13 *T-s diagram*

Power input to compressor = 526.1 × 0.1 = 52.61 kW

Example 19.10

In a 5 kW cooling capacity refrigeration system operating on a simple vapour compression cycle, the refrigerant enters the evaporator with an enthalpy of 75 kJ/kg and leaves with an enthalpy of 183 kJ/kg. The enthalpy of the refrigerant after compression is 210 kJ/kg. Show the cycle on *T-s* and *p-h*

diagrams. Calculate the following: (a) COP, (b) power input to compressor, and (c) rate of heat transfer at the condenser.

Solution

Refer to Fig. 19.14.

Mass flow rate of refrigerant,

Power input to compressor,

$$= \dot{m}(h_2 - h_1) = 0.046\ (210 - 183) = 1.24\ \text{kW}$$

Rate of heat transfer to condenser,

$$Q_r = \dot{m}(h_2 - h_4)$$
$$h_4 = h_3$$
$$Q_r = 0.046\ (210 - 75) = 6.21\ \text{kW}$$

Figure 19.14 *(a) T-s diagram, (b) p-h diagram*

Example 19.11

An ice making plant using refrigerant R-12 is having an evaporator saturation temperature of –25°C and the condenser saturation temperature of 35°C. The vapour is leaving the compressor at 65°C. Following table shows the properties of the refrigerant:

Enthalpy of superheated refrigerant at 850 kPa and 65°C = 225.5 kJ/kg

1. Calculate the coefficient of performance (COP) of this system.
2. If the capacity of the plant is 5 kW, calculate mass flow rate of refrigerant and power consumption.

Solution

Refer to Fig. 19.15.

1.
2. $\dot{m}(h_1 - h_1) = 5$

Power consumption = $\dot{m}(h_2 - h_1)$

$$= 0.0467 \times 49 = 2.288 \text{ kW}$$

Figure 19.15 *p-h diagram*

19.6 ❐ EFFECT OF SUCTION PRESSURE

The suction or evaporator pressure decreases due to the frictional resistance to the flow of refrigerant. Let the suction pressure p_s decrease to $p_{s'}$ as shown in *p-h* diagram of Fig. 19.16. The effect of decrease in suction pressure are:

1. The refrigerating effect decreases from $(h_1 - h_4)$ to $(h_{1'} - h_{4'})$
2. The compression work increases from $(h_2 - h_1)$ to $(h_{2'} - h_{1'})$

Therefore, the COP decreases for the same amount of refrigerant flow. Hence, the refrigerating capacity of the system decreases and the refrigeration cost increases.

Figure 19.16 *p-h diagram*

19.7 ❒ EFFECT OF DISCHARGE PRESSURE

The discharge (or condenser) pressure p_d increases to $p_{d'}$ due to frictional resistance to flow or refrigerant. The effects of increase in discharge pressure are (Fig. 19.17):

1. The refrigerating effect decreases from $(h_1 - h_4)$ to $(h_1 - h_{4'})$.
2. The compression work increases from $(h_2 - h_1)$ to $(h_2 - h1)$.

Figure 19.17 *p-h diagram*

Thus the COP decreases. The effect of increase in discharge pressure is similar to that of decrease in suction pressure.

19.8 ❐ EFFECT OF SUPERHEATING OF REFRIGERANT VAPOUR

The vapour compression refrigeration cycle with superheated vapour before compression on *T-s* and *p-h* diagrams are shown in Fig. 19.18. The evaporation starts at state 4 and continues upto state 1′, when it is dry and saturated. The vapours are now superheated from state 1′ to state 1, where they enter the compressor. Its effect is to increase the COP.

Figure 19.18 *Superheated vapour compression cycle: (a) T-s diagram, (b) p-h diagram*

19.8.1 Superheat Horn

Consider the *T-s* diagram for a vapour compression cycle shown in Fig 19.19. The vapours of refrigerant are wet at state 1′ and dry saturated at state 2′. The

process 1′–2′ represents wet compression. The vapours at state 1 are dry saturated and at state 2 superheated. The process 1–2 represents dry compression. The increased work of the cycle due to the replacement of wet compression by dry compression appears as the area 2–2′–2″, generally known as superheat horn.

Figure 19.19 *Wet and dry compression processes*

19.9 ❒ EFFECT OF SUBCOOLING (OR UNDERCOOLING) OF REFRIGERANT VAPOUR

Consider the *T-s* and *p-h* diagrams for the vapour compression cycle shown in Fig. 19.20 in which the refrigerant after condensation process 2′–3′, is cooled below the saturation temperature T_3' before throttling process to temperature T_3. Such a process is called

undercooling or subcooling of the refrigerant. It is generally done along the saturated liquid line. The effect of undercooling is to increase the COP.

Refrigerating effect, $R_E = h_1 - h_4 = h_1 - h_{f3}$, where $h_{f3} = h_{f3'} - c_{pf}(T_{3'} - T_3)$

Work done, $w = h_2 - h_1$

Figure 19.20 *Vapour compression cycle with subcooling of refrigerant: (a) T-s diagram, (b) p-h diagram*

Example 19.12

A food storage locker requires a refrigeration system of 42 kW capacity at an evaporator temperature of –5°C and a condenser temperature of 40°C. The refrigerant, R-12, is sub-cooled 5°C

before entering the expansion valve, and the vapour is superheated 6°C before leaving the evaporator coil. The compression of the refrigerant is reversible adiabatic. A two-cylinder vertical single acting compressor with stroke equal to 1.5 times the bore is to be used operating at 960 rpm. Determine (a) the refrigerating effect/kg, (b) the mass of refrigerant to be circulated per minute, (c) the theoretical piston displacement per minute, (d) the theoretical power, (e) the co-efficient of performance, (f) the heat removed through condenser/kg and (g) the theoretical bore and stroke of compressor.

Data for R–12 refrigerant is given below:

Solution

Given: $Q = 42$ kW, $T_1 = -5$°C or 273–5= 268 K, $T_{2'} = 40$°C or 273 + 40 = 318K, $T_{3'} - T_3 = 5$°C, $T_1 - T_{1'} = 6$°C, $n = 2$, $L = 1.5D$, $N = 960$ rpm.

The *T-s* diagram is shown in Fig. 19.21.

Figure 19.21 *T-s diagram*

Process l′–1: superheating

$$h_1 = h_{g1'} + c_{pg1'}(T_1 - T_{1'}) = 350.44 + 0.6455 \times 6$$
$$= 354.313 \text{ kJ/kg}$$
$$= 1.5778 \text{ kJ/kg.K}$$

Process 1–2: Isentropic compression

$$T_2 = 326.38 \text{ K}$$
$$h_2 = h_{g2'} + c_{pg2'}(T_2 - T_{2'})$$
$$= 368.81 + 0.6455\ (326.38 - 313)$$
$$= 379.00 \text{ kJ/kg}$$

Process 3′–3: Subcooling

$$h_2 = h_{g3'} + c_{pg3'}(T_3 - T_{3'}) = 239.03 - 1.030 \times 5 = 233.88 \text{kj/kg}$$

Process 3–4: Throttling

$$h_4 = h_3 = 233.88$$

Specific volume at suction to compressor

1. Refrigeration effect/kg, $q_{4-1} = h_1 - h_4 = 354.313 - 233.88 = 120.433$kJ/kg
2. Mass of refrigerant required/min,
3. Theoretical suction volume/min = $\dot{m} \times v_1 = 20.924 \times 0.06706 = 1.403 \text{ m}^3$/min
4. Power,
5.
6. Heat removed through the condenser/kg = $h_2 - h_3 = 379.00 - 233.88 = 145.12$ kJ/kg
7. Theoretical suction volume per cylinder per minute

19.10 ❐ VAPOUR ABSORPTION SYSTEM

The vapour absorption system uses heat energy instead of mechanical energy in order to change the conditions of the refrigerant required for the operation of the refrigeration cycle. In the vapour absorption system, the compressor is replaced by an absorber, a pump, a generator and a pressure reducing valve. The vapour refrigerant from the evaporator is drawn into the absorber where it is absorbed by the weak solution of the refrigerant forming a strong solution. This strong solution is pumped to the generator where it is heated by some external source like steam or heating oil. During the heating process, the vapour refrigerant is driven off by the solution and enters into the

condenser where it is liquefied. The liquid refrigerant then flows into the evaporator, thereby completing the cycle. A line diagram of vapour absorption refrigeration system is shown in Fig. 19.22.

Figure 19.22 *Vapour absorption refrigeration system*

19.11 ❒ WORKING PRINCIPLE OF VAPOUR ABSORPTION REFRIGERATION SYSTEM

The schematic diagram of vapour absorption system is shown in Fig. 19.23. It consists of

1. an absorber,
2. a pump,
3. a generator and
4. a pressure reducing valve

    The function of the pressure reducing valve is to replace the compressor of vapour compression system. The other components of the system are

5. condenser,
6. receiver,
7. expansion valve and
8. evaporator as in the vapour compression system.

Figure 19.23 *Schematic diagram for vapour absorption system*

19.11.1 Working

In the vapour absorption system, the low pressure ammonia vapour leaving the evaporator, enters the absorber where it is absorbed by the cold water in the absorber. The solution of ammonia vapour in water is called aqua-ammonia. The absorption of ammonia vapour in water lowers the pressure in the absorber which in turn draws more ammonia vapour from the evaporator. As a result of this, the temperature of solution rises. Some cooling arrangement (usually water cooling) is employed in the absorber to remove the heat of solution evolved there. This increases the absorption capacity of water. The strong solution thus formed

in the absorber is pumped to the generator by the liquid pump.

The strong ammonia solution in the generator is heated by some external source such as hot gas, steam or heating oil. During the heating process, the ammonia vapour is driven off the solution at high pressure leaving behind the hot weak solution of ammonia in the generator. This weak ammonia solution flows back to the absorber at low pressure after passing through the pressure reducing valve. The high pressure ammonia vapour from the generator is condensed in the condenser to a high pressure liquid ammonia. This liquid ammonia is passed to the expansion valve through the receiver

and then to the evaporator. This completes the vapour absorption cycle.

19.12 ❐ ADVANTAGES OF VAPOUR ABSORPTION SYSTEM OVER VAPOUR COMPRESSION SYSTEM

The advantages of vapour absorption system over vapour compression system are listed in Table 19.1.

Table 19.1 *Comparison of vapour absorption and vapour compression systems*

19.13 ❐ COEFFICIENT OF PERFORMANCE OF AN IDEAL VAPOUR ABSORPTION SYSTEM

The ideal vapour absorption refrigeration system is shown in Fig. 19.22

Let Q_g = heat given to the refrigerant in the generator.

Q_{rc} = heat rejected from the condenses to the atmosphere.

Q_{ra} = heat rejected from the absorber to the atmosphere.

Q_e = heat absorbed by the refrigerant in the evaporator.

Q_p = heat added to the refrigerant due to pump work.

$Q_r = Q_{rc} + Q_{ra}$

= total heat rejected from the condenser and absorber to atmosphere

For heat balance, neglecting Q_p according to the first law of thermodynamics, we have

$$Q_r = Q_{rc} + Q_{ra} = Q_g + Q_c$$

Let T_g = temperature at which heat is given to the generator.

T_r = temperature at which heat is rejected to atmosphere or cooling water from the condenser and absorber.

T_e = temperature at which heat is absorbed in the evaporator.

The vapour absorption system can be considered as a perfectly reversible system. Therefore, the initial entropy of

the system must be equal to the entropy of the system after the change in its conditions.

Maximum coefficient of performance,

Neglecting pump work,

If T_a = temperature at which heat is rejected from the absorber

T_c = temperature at which heat is rejected from the condenser.

Example 19.13

A geothermal well at 130°C supplies heat at a rate of 100,500 kJ/

h to an absorption refrigeration system. The environment is at 30°C and the refrigerated space is maintained at –22°C. Determine the maximum possible heat removal from the refrigerated space.

Solution

Given: $T_r = 273 + 35 = 308$ K, $T_g = 273 + 110 = 383$ K, $T_e = 273 - 5 = 268$ K

Example 19.14

In an absorption type refrigerator, the heat is supplied to ammonia generator by condensing steam at

2.5 bar and dryness fraction 0.9. The temperature in the refrigerator is to be maintained at –5°C. Calculate the maximum COP possible.

If the refrigeration load is 20 tonnes and actual COP is 75% of the maximum COP, find the mass of steam required per hour. Take ambient temperature as 30°C.

Solution

Given: $p = 2.5$ bar, $x = 0.9$, $T_e = 273 - 5 = 268$ K, $Q = 20$TR, $(COP)_{max} = 0.75(COP)_{max}$, $T_r = 273 + 30 = 303$ K, From steam tables $h_f = 535.34$ kJ/kg. $h_g = 2716.9$ kJ/kg.

Saturation temperature of steam at $p = 2.5$ bar, $T_s = 273 + 127.43 = 400.43$ K $= T_g$

$$(COP)_{act} = 0.75 \times 1.863 = 1.397$$

Actual heat supplied =

Latent heat of steam, $h_{fg} = h_g - h_f = 2716.9 - 535.35 = 2181.56$ kJ/kg

Mass of steam required per hour,

19.14 ❒ AMMONIA-WATER (OR PRACTICAL) VAPOUR ABSORPTION SYSTEM (nh$_3$ – h$_2$o)

In the $NH_3 - H_2O$ vapour absorption system, ammonia is the refrigerant and water is the absorber. An analyser, a rectifier and two heat exchangers are added to the simple vapour absorption

system. The complete schematic diagram of ammonia-water absorption system is shown in Fig. 19.24.

The functions of various accessories are:

1. **Analyser:** In the simple vapour absorption system, some water is also varporised along with ammonia and flows into the condenser. These water vapours further move and enter into the expansion valve, where they freeze and choke the pipeline. The analyser is used to remove these water vapours. The analyser may be an integral part of the generator or as a separate piece of equipment. It consists of a series of trays mounted above the generator. The strong solution from the absorber and the aqua from the rectifier (or dehydrator) are introduced at the top of the analyser and flows downward on the trays and into the generator. In this way, considerable liquid surface area is exposed to the vapour rising from the generator. The vapours are cooled and most of the water vapours condense, so that mainly ammonia vapours leave the top of the analyser. This also reduces the external heat required in the generator since aqua is heated by the vapour.
2. **Rectifier or Dehydrators:** The function of the rectifier is to completely remove the water vapours still left in the analyser. The rectifier is a closed type vapour cooler of the double pipe type, shell and coil, or shell and tube type. Its function is to further cool the ammonia vapours leaving the analyser so that the remaining water vapours are condensed. The condensate from the rectifier is returned to the top of the analyser by a drip return pipe.
3. **Heat exchangers:** The heat exchanger-1 provided between the pump and the generator is used to cool the weak hot solution returning from the generator to the absorber. The heat removed from the weak solution raises the temperature of the strong solution leaving the pump and going to analyser and generator. This operation reduces the heat supplied to the generator and the amount of cooling required for the absorber.

Figure 19.24 *Ammonia-water vapour absorption system*

The heat exchanger-2 provided between the condenser and the evaporator may be called liquid sub-cooler. The liquid refrigerant leaving the condenser is sub cooled by the low temperature ammonia vapour from the evaporator. This sub-cooled liquid is now passed to the expansion valve and then to the evaporator.

Net refrigerating effect, R_E = Heat absorbed in the evaporator

Total energy supplied to the system,

$$W_t = \text{Work done by the pump + Heat supplied in generator}$$

19.15 ❒ LITHIUM BROMIDE-WATER VAPOUR ABSORPTION SYSTEM (LiBr-H_2O)

In the LiBr-water system, water is the refrigerant and LiBr the absorber. The lithium bromide solution has a strong affinity for water vapour because of its very low vapour pressure. The lithium bromide-water vapour absorption system is shown in Fig. 19.25. The

evaporator and absorber are placed in one shell which operates at the same low pressure of the system. The generator and condenser are placed in another shell which operates at the same high pressure of the system.

Figure 19.25 *Lithium bromide-water vapour absorption system*

19.15.1 Working Principle

The water for process requirements is chilled as it is pumped through the chilled-water tubes in the evaporator by giving up heat to the refrigerant water sprayed over the tubes. Since the pressure inside the evaporator is maintained very low, therefore, the refrigerant water evaporates. The water vapours thus formed are absorbed by the strong lithium bromide solution which is

sprayed in the absorber. In absorbing the water vapour, the lithium bromide solution helps in maintaining very low pressure needed in the evaporator, and the solution becomes weak. This weak solution is pumped by a pump to the generator where it is heated up by using steam in the heating coils. A portion of water is evaporated by the heat and the solution becomes more strong. This strong solution is passed through the heat exchanger and then sprayed in the absorber. The weak solution of lithium bromide from the absorber to the generator is also passed through the heat exchanger. This weak solution gets heat from the strong solution in the heat exchanger, thus reducing the quantity of steam required to heat the weak solution

in the generator.

The refrigerant water vapours formed in the generated due to healing of solution are passed to the condenser where they are cooled and condensed by the cooling water flowing through the condenser water tubes. The cooling water for condensing is pumped from the cooling water pond. This cooling water first enters the absorber where it takes away the heat of condensation and dilution. The condensate from the condenser is supplied to the evaporator to compensate the water vapor formed in the evaporator. The pressure reducing valve reduces the pressure of condensate from the condenser pressure to the evaporator pressure. The cooled water

from the evaporator is pumped and sprayed in the evaporator in order to the cool the water flowing through the chilled tubes. This completes the cycle.

19.15.2 Lithium Bromide-Water System Equipment

A line diagram of two-shell lithium bromide-water vapour absorption refrigeration system is shown in Fig. 19.26. The various components are:

Figure 19.26 *Lithium bromide-water vapour absorption system*

1. **Generator:** It consists of tube bundles submerged in the solution heated by steam or hot liquids.
2. **Condenser:** It. consists of tube bundles located in the vapour space over the generator shielded from carry over of salt by eliminators. Cooling water to condenser first passes through the absorber.
3. **Absorber:** It consists of bundles over which the strong absorbent is sprayed. Refrigerant vapours are condensed into the absorbent releasing heat to the cooling water passing through it.
4. **Evaporator:** Evaporators are tube bundles over which the refrigerant water is sprayed and evaporated. The liquid to be cooled passes inside the tubes.
5. **Solution heat exchangers:** Solution heat exchangers are all of steel shell and tube construction.
6. **Purgers:** All units include a purger which is used to remove non-condensable gases. Non-condensable gases present in

small quantities can raise the total pressure in the absorber sufficiently to significantly change the evaporator pressure. Very small pressure increases cause appreciable change in the refrigerant evaporating temperatures.

7. **Expansion Device:** Mechanical expansion valves are not used in absorption units. The flow or refrigerant liquid to evaporator is controlled by an orifice or other fixed restriction between the condenser and the evaporators.
8. **Pump**: Other details about pumps etc., are shown in Fig. 19.26.

Capacity control: All lithium bromide-water cycle absorption machines meet load variations and maintained chilled water temperature control by varying the rate of re-concentration of the absorbent solution.

At any given constant load, the chilled water temperature is maintained by a temperature difference between refrigerant and chilled water. The refrigerant temperature is maintained in turn by absorber being supplied with a flow rate and concentration of solution,

and by absorber cooling water temperature.

Load changes are reflected by corresponding changes in chilled water temperature. Aload reduction, for example, results in less temperature difference being required in the evaporator and a reduced requirement for solution flow or concentration. The resultant chilled water temperature drop is met basically by adjusting the rate of re-concentration to match the reduced requirements of the absorber.

All units sense the chilled water temperature changes resulting from a load change with a thermostat in the leaving chilled water.

19.16 ❒ COMPARISON OF AMMONIA-WATER AND LITHIUM BROMIDE-WATER ABSORPTION SYSTEMS

The comparison of ammonia-water and lithium bromide-water systems are given m Table 19.2

Table 19.2 *Comparison of ammonia-water and lithium bromide-water systems*

Exercises

19.1. The evaporator and condenser temperatures in an NH_3 refrigeration system are –10°C and 40°C respectively. Determine per TR basis: (a) mass flow rate; (b) compressor work, (e) condenser heat rejection; (d) C.O.P.; and (e) refrigerating efficiency. Use only the propertiesof NH_3 given below:

For superheated NH_3 at 15.55 bar, the following values may be taken

[**Ans.** 0.198 kg/min: 0.82 kW; –4. 32 kW; 4.27; 81.2%]

19.2. In a vapour compression refrigeration system using *R*-12, the evaporator pressure is 1.4 bar and the condenser pressure is 8 bar. The refrigerant leaves the condenser sub-cooled to 30°C. The vapour leaving the evaporator is dry and saturated. The compression process is isentropic. The amount of heat rejected in the condenser is 13.42 MJ/min. Determine: (a) refrigerating effect in kJ/kg; (b) refrigerating load in TR; (c) compressor input in kW; and (d) COP.

[**Ans.** 114 k//kg; 49 TR: 51.4 kW; 3.35]

19.3. A vapour compression refrigerator works between the temperature limits of –20°C and 25°C. The refrigerant leaves the compressor in dry saturated condition. If the liquid refrigerant is undercooled to 20°C before entering the throttle valve determine:

1. work required to drive the compressor ;
2. refrigerating effect produced per kg of the refrigerant; and
3. theoretical COP.

Assume specific heat of the refrigerant as 4.8. The properties of the refrigerant are

[**Ans.** 189.7 kJ/kg; 990.2 kJ/kg; 5.01]

19.4. A food storage chamber requires a refrigeration system of 12 TR capacity with an evaporator temperature of –8°C and condenser temperature of 30°C. The

refrigerant *R*-12 is sub-cooled by 5°C before entering the throttle valve, and the vapour is superheated by 6°C before entering the compressor. If the liquid and vapour specific heats are 1.235 and 0.733 kJ/kg.K respectively, find: (a) refrigerating effect per kg; (b) mass of refrigerant circulated per minute; and (c) coefficient of performance.

The relevant properties of the refrigerant *R*-12 are given below:

[**Ans.** 130.05 kJ/kg; 19.4 kg; 6.2]

19.5. An ammonia refrigerator produces 30 tonnes of ice form and at 0°C in 24 hours. The temperature range of the compressor is from 25°C to –15°C. The vapour is dry saturated at the end of

compression and an expansion valve is used. Assume a coefficient of performance to be 60% of the theoretical value. Calculate the power required to drive the compressor. Latent heat of ice = 335 kJ/kg. Properties of ammonia are.

[**Ans.** 33.24 kW]

19.6. A freezer of 20 TR capacity has evaporator and condenser temperature of –30°C and 25°C respectively. The refrigerant *R*-12 is sub-cooled by 4°C before it enters the expansion valve and is superheated by 5°C before leaving the evaporator. The compression is isentropic and the valve throttling and clearance are to be neglected. If a six cylinder, single acting compressor with

stroke equal to bore running at 1000 rpm is used, determine (a) COP, of the refrigerating system, (b) mass of refrigerant to be circulated per min, (c) theoretical piston displacement per minute, and (d) theoretical bore and stroke of the compressor. The specific heat of liquid *R*-12 is 1.235kJ/kg K and of vapour *R*-12 is 0.733 kJ/kg.K. The properties of *R*-12 are given below:

[**Ans.** 3.64; 34.12 kg/min; 5.56 m/min; 0.106 m]

19.7. A refrigeration plant of 8 TR capacity has its evaporation temperature of – 8°C and condenser temperature of 30°C. The refrigerant is sub-cooled by 5°C before entering into the expansion valve and vapour is superheated by 6°C before leaving the refrigerator. The

suction pressure drop is 0.2 bar in the suction valve and discharge pressure drop is 0.1 bar in the discharge valve.

If the refrigerant used is *R*-12, find out the COP, of the plant and theoretical power required for the compressor. Assume compression is isentropic. Use *p-h* chart for calculation.

19.8. An ammonia refrigerator works between – 6.7°C and 26.7°C, the vapour being dry it the end of isentropic compression. There is no under cooling of liquid ammonia and the liquid is expanded through a throttle valve after leaving the condenser. Sketch the cycle on the *T-s* and *p-h* diagram and calculate the refrigeration effect per kg ammonia and the theoretical coefficient of

performance of the unit with the help of the properties given below:

[**Ans.** 1028.3 kJ/kg, 7.2]

19.9. An ammonia refrigerating machine fitted with an expansion valve works between the temperature limits of −10°C and 30°C. The vapour is 95%, dry at the end of isentropic compression and the fluid leaving the condenser is at 30°C If the actual coefficient of performance is 60% of the theoretical, find the ice produced per kW hour at 0°C from water at 10°C. The latent heat of tee is 335 kJ/kg. The ammonia has the following properties:

[**Ans.** 33.24 kg/kWh]

19.10. A *R-12* refrigerating machine

works on vapour-compression cycle. The temperature of refrigerant in the evaporator is – 20°C. The vapour is dry saturated when it enters the compressor and leaves it in a superheated condition. The condenser temperature is 30°C. Assuming specific heat at constant pressure for *R*-12 in the superheated condition as 1,884 kJ/kg.K, determine:

1. condition of vapour at the entrance to the condenser:
2. condition of vapour at the entrance to the evaporator; and
3. theoretical COP of the machine. The properties of *R*-12 are:

[**Ans.** 33.8°C;29% dry; 4.07]

19.11. A CO_2 refrigerating plant fitted with an expansion valve, works between the pressure limits of 54.81 bar and 20.93 bar. The vapour is compressed isentropically and leaves the compressor cylinder at 32°C. The condensation

takes place at 18°C in the condenser and there is no undercooling of the liquid. Determine the theoretical coefficient of performance of the plant. The properties of CO_2 are:

[**Ans.** 4.92]

19.12. A single stage NH_3 refrigeration system has cooling capacity of 500 kW. The evaporator and condenser temperatures are –10°C and 30°C respectively. Assuming saturation cycle, determine: (a) mass flow rate of refrigerant; (b) adiabatic discharge temperature, (c) compressor work in kW; (d) condenser heat rejection, (e) COP; and (f) compressor swept volume in m^3/min, if volumetric efficiency is 70%.

The following values may be taken:

The properties of superheated NH_3 at condenser pressure of 1 1.66 bar (30°C) are as follows:

At 85°C, h = 1621.8 kJ/kg; s = 5.5484 kJ/kg.K.

At 90°C, h = 1634.5 kJ/kg;

s = 5.4838 kJ/kg.K;

[**Ans.** 0.45 kg/s; 88.3°C; 89.5 kW; 590 kW; 5.585; 16.2 m^3/min]

19.13. A 15 TR Freon 22 vapour compression system operates between a condenser temperature of 40°C and an evaporator temperature of 5°C.

1. Determine the compressor discharge temperature:
    1. Using the *p-h* diagram for Freon 22.
    2. Using saturation properties of Freon 22 and assuming the specific heat of its vapour as 0.8 kJ/kg.K.
    3. Using superheat tables for Freon 22.
2. Calculate the theoretical piston displacement and power consumption of the compressor per ton of refrigeration.

19.14. A simple saturation ammonia compression system has a high pressure of 1.35 MN/m^2 and a low pressure of 0.19 MN/m^2. Find per 400,000 kJ/h of refrigerating capacity, the power consumption of the compressor and COP of the cycle.

19.15.

1. A Freon 22 refrigerating machine operates between a condenser temperature of 40°C and an evaporator temperature of 5°C. Calculate the increase (per cent) in the theoretical piston displacement and the power consumption of the cycle:
    1. If the evaporator temperature is reduced to 0°C,
    2. If the condenser temperature is increased to 45°C.
2. Why is the performance of a vapour compression machine more sensitive to change in evaporator temperature than to an equal change in the condenser temperature?

19.16. In a vapour compression cycle

saturated liquid Refrigerant 22 leaving the condenser at 40°C is required to expand to the evaporator temperature of 0°C in a cold storage plant.

1. Determine the percentage saving in network of the cycle per kg of the refrigerantif an isentropic expander could be used to expand the refrigerant in place of thethrottling device.
2. Also determine the percentage increase in refrigerating effect per kg of refrigerant as a result of use of the expander. Assume that compression is isentropic from saturated vapour state at 0°C to the condenser pressure.

Chapter 20

Air-Conditioning and Psychrometrics

20.1 ❐ INTRODUCTION

The air-conditioning is that branch of engineering which deals with the study of supplying and maintaining desirable internal room atmospheric conditions of air for human comfort. The four important factors for human comfort are: temperature, humidity, purity and motion of air.

The working substance for air-conditioning is moist air, which a mixture of two gases. One of these is dry air, which itself is a mixture of a

number of gases and the other is water vapour which may exist in a saturated or superheated state. Moist air is not a pure substance in any process in which condensation or evaporation of moisture occurs. Both dry air and water vapour can be considered as perfect gases since both exist in the atmosphere at low pressures. Hence, perfect gas laws to both and Dalton's law of partial pressures for non-reactive mixture of gases can be applied to dry air part only.

20.2 ❐ PRINCIPLES OF PSYCHROMETRY

The psychrometry is that branch of engineering which deals with the study of moist air i.e. dry air mixed with water vapour.

Dry air: The dry air consists of O_2 =

0.2099, N_2 = 0.7803, Ar = 0.0094, CO_2 = 0.0003 and H_2 = 0.0001 parts by volume.

Molecular weight of air = 28.966

Gas constant, R_a = 0.287 kJ/kg K

Density of air = 1.293 kg/m^3 at 1.01325 bar and 0°C

Specific heat of air at constant pressure = 1.005 kJ/kg K

Molecular weight of water (vapour) = 18.02

Gas constant for water vapour, R_v = 0.461 kJ/kg K

Barometric pressure = 1.01325 bar or 760 mm Hg

Dalton's law of partial pressures: The Dalton's law of partial pressures states that for a mixture of ideal gases, the total pressure is equal to the sum of the partial pressures, which each constituent would exert if it occupied the whole space alone. Thus,

Barometric pressure of mixture of dry air and water vapour is given by

where p_a = partial pressure of dry air

p_v = partial pressure of water vapour

Moist air: It is a mixture of dry air and water vapour.

Saturated air: It is a mixture of dry air and water vapour when air has absorbed maximum amount of water vapour. At saturation $p_v = p_s$, the saturation pressure.

Degree of saturation (μ):

Specific humidity or humidity ratio w: It is the mass of water vapour present in one kg of dry air. It is expressed in terms of kg per kg of dry air (kg/kg d.a)

Absolute humidity: It is the mass of water vapour present in 1m^3 of dry air. It is expressed in terms of kg per cubic metre of dry air (kg/m^3 d.a.)

Relative humidity (RH or ϕ): It is the

ratio of actual mass of water vapour in a given volume of moist air to the mass of water vapour in the same volume of saturated air at the same temperature and pressure.

Dry Bulb Temperature (DBT, t_d): It is the temperature of air recorded by a thermometer, when it is not affected by the moisture present in the air.

Wet Bulb Temperature (WBT, t_w): It is the temperature of air recorded by a thermometer, when its bulb is surrounded by a wet cloth exposed to the air.

The DBT and WBT are measured by a Sling Psychrometer, as shown in Fig. 20.1.

Wet Bulb Depression, WBD = DBT – WBT

Dew Point Temperature (DPT, t_{dp}): It is the temperature of air recorded by a thermometer, when the water vapour present in it begins to condense. Thus, DPT is the saturation temperature (t_{sat}) corresponding to the partial pressure of water vapour p_v

Dew Point Depression, DPD = DBT – DPT

Figure 20.1 *Sling psychrometer*

20.3 ❐ PSYCHROMETRIC RELATIONS

1. **Specific (or absolute) humidity, humidity ratio or moisture content, *w*:** It is defined as the ratio of the mass of water vapour to the mass of dry air in a given volume of the mixture.

   Let p, V, T, m = pressure, volume, absolute temperature and mass respectively

R, M, v, $\bar{R}$ = gas constant, molecular mass, specific volume, and universal gas constant respectively

Subscripts a, v = air and vapour respectively.

For air,

And

For water vapour,

Substituting for m_v and m_a from these equations in Eq. (20.2), we get

Now actual total pressure, $p_b = p_a + p_v$

For $m_a = 1$ kg, $m_v = w$ kg and total mass of moist air

For saturated air, $p_v = p_s$ and maximum humidity ratio

where p_s = partial pressure of air corresponding to saturation temperature i.e. t_d.

2. **Degree of saturation or percentage humidity, μ:** The ratio of actual specific humidity w to the specific humidity w_s of saturated air at the same temperature is termed as the degree of

saturation.

The degree of saturation is a measure of the capacity of air to absorb moisture.

3. **Relative Humidity (RH or ϕ):** It is defined as the ratio of mass of water vapour m_v in a certain volume of moist air at a given temperature to the mass of water vapour m_{vs} in the same volume of saturated air at the same temperature.

   For a perfect gas $p_v v_v = p_s v_s$

   Now

4. **Pressure of water vapour:**
   1. The partial pressure of water vapour, according to Carrier's equation, is given by

      where p_w = saturation pressure corresponding to WBT

      p_b = barometric pressure

      t_d = DBT

      t_w = WBT

   2. Modified Apjohn equation:
   3. Modified Ferrel equation:
5. **Vapour density or Absolute humidity:**

   Let p_v = density of water vapour corresponding to its partial pressure and DBT, t_d.

   p_a = density of dry air.

Then $m_v = V_v\rho_v$ and $m_a = V_a\rho_a$

Since $V_a = V_v$, therefore,

Or $\rho_v = w\rho_a$

Now $p_a\ V_a = m_aR_aT_d$

For $m_a = 1$ kg,

20.4 ❒ ENTHALPY OF MOIST AIR

The temperature-entropy diagram is shown in Fig. 20.2.

Enthalpy of moist air = Enthalpy of dry air + Enthalpy of water vapour associated with dry air

$$h = h_a + wh_v \text{ per kg of dry air}$$

where $h_a = c_{pa}\ t_d = 1.005\ t_d$ kJ/kg

$c_{pa} = 1.005$ kJ/kg. K
t_d = DBT of air in °C.
$h_v = c_{pw}\ t_{dp} + [(h_{fg})_{dp} + c_{pv}\ (t_d - t_{dp})]$kJ/kg

$$= 4.1868t_{dp} + (h_{fg})_{dp} + 1.88(t_d - t_{dp})$$

where $c_{pw} = 4.1868$ kJ/kg K is the specific heat of liquid water

$(h_{fg})_{dp}$ = latent heat of vaporization at DPT
$c_{pv} = 1.88$ kJ/kg.K is the specific heat of superheated water vapour

Figure 20.2 *Temperature-entropy diagram*

t_{dp} = DPT, °C
w = specific humidity

$$h = 1.005\ t_d + w\ [4.1868\ t_{dp} + (h_{fg})_{dp} + 1.88\ (t_d + t_{dp})]$$

20.5 ❐ HUMID SPECIFIC HEAT

Eq. (20.16) can be written as,

$$h = (c_{pa} + wc_{pv})t_d + w(h_{fg})_{0°C}$$

is termed as the humid specific heat. It is the specific heal of moist air $(1 + w)$ kg per kg of dry air.

At low temperatures of air-conditioning the value of w is very small. An approximate value of c_p of 1.0216 kJ/(kg d.a.).K may be taken for nil practical

purposes

Example 20.1

A mixture of dry air and water vapour is at a temperature of 20°C under a total pressure of 735 mm Hg. The dew point temperature is 15°C. Calculate the following:

1. Partial pressure of water vapour,
2. Relative humidity,
3. Specific humidity,
4. Specific enthalpy of water vapour
5. Enthalpy of air per kg of dry air, and
6. Specific volume of air per kg of dry air.

Solution

1. From steam tables, the partial pressure of water vapour at t_{dp} = 15°C

$$p_v = 12.79 \text{mm Hg}$$

Now 760mm Hg = 1.01325×10^5 N/m^2

$$1 \text{mm Hg} = 133.32 \text{ N/m}^2$$

$$\therefore \quad p_v = 12.79 \times 133.32 = 1705.16 \text{N/m}^2$$

2. Saturation pressure of water vapour at 20°C *DBT*

$$p_s = 17.54 \text{mm Hg} = 17.54 \times 133.32 = 2338.4 \text{ N/m}^2$$

Relative humidity,

3. Specific humidity, w =
4. Latent heat of vaporization of water at $t_d = 20°C$, $(h_{fg})_{20°C}$ = 2454.1kJ/kg

Latent heat of vaporization of water at t_{dp} $=15°C$, $(h_{fg})_{15°C} = 2465.9$ kJ/kg

Specific enthalpy of water vapour,

$$h_v = [4.1868 t_{dp} + (h_{fg})_{dp} + 1.88(t_d - t_{dp})]$$

$$= [4.1868 \times 15 + 2465.9 + 1.88(20 - 15)]$$

$$= 2538.1 \text{ kJ/kg } w.v$$

5. Enthalpy of air per kg d.a., $h = 1.005\ t_d + wh_v$

$$= 1.005 \times 20 + 0.011 \times 2538.1$$

$$= 48.02 \text{ kJ/kg d.a.}$$

6. Specific volume of air per kg of d.a.

$$= 0.873 \text{ m}^3\text{/kg d.a}$$

Example 20.2

A sling psychrometer gave the following readings:

DBT = 30°C, WBT = 20°C,
Barometer reading = 740 mm of Hg.

Determine: (a) dew point temperature, (b) relative humidity, (c) specific humidity (d) degree of saturation, (e) vapour density, and (f) enthalpy of mixture per kg of dry air

Solution

Given: $t_d = 30°C$, $t_w = 20°C$, $p_b =$ 740 mm Hg

1. Corresponding to $t_w = 20°C$, from steam tables, saturation pressure, $p_w = 0.023385$ bar

   Barometric pressure, $p_b = 740 \times 133.32 \times 10^{-5}$
   $= 0.986568$ bar

   Partial pressure of water vapour

   Corresponding to $p_v = 0.01703$ bar pressure,

from steam tables, dew point temperature

$$t_{dp} = 15°C$$

2. Corresponding to $t_d = 30°C$, from steam tables, saturation pressure of water vapour,

$$p_s = 0.04246 \text{ bar}$$

3. Specific humidity,

$$= 0.010925 \text{ kg/kg d.a}$$

4. Specific humidity of saturated air,

Degree of saturation,

5. Vapour density,

$$= 0.01218 \text{ kg/m}^3 \text{ of d.a.}$$

6. From steam tables, $(h_{fg})_{dp} = {}_{15°C} = 2465.9$ kJ/kg

Enthalpy of mixture per kg d.a.,

$$h = 1.0216\, t_d + w[(h_{fg})_{dp} + 2.3068\, t_{dp}]$$
$$= 1.0216 \times 30 + 0.010925\, [2465.9 + 2.3068 \times 15] = 57.966 \text{ kJ/kg d.a.}$$

Example 20.3

A room 5m × 4m × 3m contains moist air at 40°C. The barometric pressure is 1 bar and relative humidity is 70%. Determine: (*a*) humidity ratio, (*b*) dew point temperature, (*c*) mass of dry air, and

(*d*) mass of water vapour.

If the moist air is further cooled at constant pressure until the temperature is 15°C, find the amount of water vapour condensed.

Solution

Given: V = 5 × 4 × 3 = 60 m^3, t_d = 40°C, p_b = 1 bar, ϕ = 0.70.

1. Saturation pressure of vapour corresponding to t_d = 40°C, from steam tables is,

   $$p_s = 7.384 \text{ kPa or } 0.07384 \text{ bar}$$
   $$p_v = 0.7 \times 0.07384 = 0.05169 \text{ bar}$$

   Humidity ratio,

   $$= 0.0339 \text{ kg/kg d.a or } 33.9 \text{ g/kg d.a}$$

2. DPT, t_{dp} = saturation temperature corresponding to p_v = 0.05169 bar or 5.169 kPa

   $$= 33.46°\text{C from steam tables}$$

3. $p_a = p_b - p_v = 1.0 - 0.05169 = 0.94831$ bar

   Mass of dry air,

4. Now w =

Mass of water vapour, $m_v = 0.0339 \times 63.34 =$ 2.147 kg

Saturation pressure corresponding to 15°C, from steam tables is,

$$p_s = p_v = 1.707 \text{ kPa or } 0.01707 \text{ bar}$$

Now $p_a = p_b - p_v = 1 - 0.01707 = 0.98293$ bar or 98.293 kPa

Mass of dry air,

Mass of water vapour, $m_v = wm_a = 0.0108 \times 71.35 = 0.77$ kg

Amount of water vapour condensed = 2.147 − 0.77 = 1.377 kg

20.6 ❐ THERMODYNAMIC WET BULB TEMPERATURE OR ADIABATIC SATURATION TEMPERATURE (AST)

The thermodynamic wet bulb temperature or AST is the temperature at which the air can be brought to saturation state adiabatically by the evaporation of water in the flowing air.

A schematic representation of this process, called the adiabatic saturation process, is shown in Fig. 20.3. It consists of an adiabatic enclosure containing adequate quantity of water. There is also an arrangement for feed water from the top.

The unsaturated air enters the enclosure at section 1. As the air flows through the enclosure water evaporates which is carried by the flowing stream of air. Thus the specific humidity of air increases. The water level in the enclosure is maintained by the feed water. Both the air and water are cooled as the evaporation takes place. The process continues until the energy transferred from the air to water is equal

to the energy required to vaporise water. When steady conditions are reached, the air flowing at section 2 becomes saturated with water vapour. The temperature of the saturated air at section 2 is known as thermodynamic wet bulb temperature or adiabatic saturation temperature. The adiabatic saturation process is represented on *T*-s diagram as shown by the curve 1-2 in Fig. 20.4. The adiabatic saturation temperature is taken equal to the wet bulb temperature.

Let t_{d1} = DBT of initial unsaturated air

t_{d2} = DBT of saturated air

= adiabatic saturation temperature, t_w

h_1, h_2 = enthalpy of unsaturated and saturated air respectively

w_1, w_2 = specific humidity of air at sections 1 and 2 respectively

h_{fw} = sensible heat of water at adiabatic saturation temperature

For energy balance of air at inlet and outlet, we have

$$h_1 + (w_2 - w_1)h_{fw} = h_2$$

or $h_1 + wh_{fw1} = h_2 + wh_{fw1}$

Now $h_1 = h_{a1} + w_1\ h_{s1}$

and $h_2 = h_{a2} + w_2\ h_{s2}$

where h_{a1} = enthalpy of 1 kg dry air at

DBT = $t_{d1} = c_{pa}\, t_{d1}$

h_{s1} = enthalpy of superheated vapour at t_{d1} per kg of vapour

h_{a2} = enthalpy of 1 kg dry air at WBT = $t_w = c_{pa}\, t_{d2}$

h_{s2} = enthalpy of saturated vapour at WBT = t_w per kg of vapour

Thus, we get

$$(h_{a1} + w_1 h_{s1}) - w_1 h_{fw} = (h_{a2} + w_2 h_{s2}) - w_2 h_{fw}$$
$$w_1(h_{s1} + h_{fw}) = w_2\,(h_{s2} + h_{fw}) + (h_{a2} - h_{a1})$$

Also $h_1 = c_{pa} t_{d1} + w_1 h_{s1}$

$$h_2 = c_{pa} t_{d2} + w_2 h_{s2}$$

Figure 20.3 *Adiabatic saturator*

Figure 20.4 *T-s diagram for adiabatic saturation process*

where $h_{s2} = h_{f2} + h_{fg2}$

$$h_{s1} = h_{s2} + c_{pv}\,(t_{d1} - t_{d2}) = h_{f2} + h_{fg2} + c_{pv}\,(t_{d1} - t_{d2})$$

From the energy balance, we get

$$c_{pa}t_{d1} + w_1[h_{f2} + h_{fg2} + c_{pv}\,(t_{d1} - t_{d2})] + (w_2 - w_1)\,h_{f2} = c_{pa}t_{d2} + w_2\,(h_{f2} + h_{fg2})$$

or $(c_{pa} + w_1\,c_{pv})t_{d1} - (c_{pa} + w_2\,c_{pv})t_{d2} = h_{fg2}\,(w_2 - w_1)$

where c_{p1} and c_{p2} are humid specific heats

If $c_{p1} = c_{p2} = c_p$, then

$$c_p\,(t_{d1} - t_{d2}) = h_{fg2}\,(w_2 - w_1)$$

Example 20.4

Atmospheric air at 1 bar enters the adiabatic saturator. The dry bulb temperature is 30°C and wet bulb temperature 20°C during adiabatic

saturation process. Calculate the following:

1. Humidity ratio of entering air,
2. Vapour pressure and relative humidity at 30°C, and
3. Dew point temperature.

Solution

Given: p_b = 1 bar, t_d = 30°C, t_w = 20°C

1. Saturation pressure of vapour at 20°C, from steam tables is,

   p_{v2} = 2.34 kPa or 0.0234 bar

   Enthalpy of saturated vapour at 20°C, $h_{s2} = h_{g2}$ = 2538.2 kJ/kg

   Sensible heat of water at 20°C, h_{fw} = 83.9 kJ/kg

   Enthalpy of saturated vapour at 30°C, $h_{s1} = h_{g1}$ = 2556.4 kJ/kg

   Enthalpy of 1 kg unsaturated air at t_d = 30°C,

   $$h_{a1} = mc_{pa}t_d = 1\times1.005\times30 = 30.15\text{kJ/kg}$$

   Enthalpy of 1 kg saturated air at t_w = 20°C,

$$h_{a2} = mc_{pa}t_w = 1 \times 1.005 \times 20 = 20.10 \text{ kJ/kg}$$

2. Now

$$0.0107 - 0.0107\, p_{v1} = 0.622\, p_{v1}$$

$$p_{v1} = 0.0169 \text{ bar}$$

Saturation pressure corresponding to 30°C from steam tables is,

$$p_s = 4.246 \text{ kPa or } 0.04246 \text{ bar}$$

Relative humidity,

3. Dew point temperature is the saturation temperature corresponding to the partial pressure of water vapour, p_{v1} = 0.0169 bar or 1.69 kPa

From steam tables, we have

$$t_{dp} = 14.85°\text{C}$$

20.7 ❐ PSYCHROMETRIC CHART

The psychrometric chart is a graphical representation of the complete thermodynamic properties of moist air and psychrometric analysis of air-conditioning processes. The chart most commonly used is the w-t_d chart, which

specifies specific humidity (ω) or water vapour pressure (p_v) along the ordinate and the dry bulb temperature (t_d) along the abscissa. The chart is normally constructed for a standard atmospheric pressure of 760 mm Hg or 1.01325 bar corresponding to the pressure at the mean sea level. A typical layout of this chart is shown in Fig. 20.5.

Some of the important lines on the psychrometic chart are explained below:

DBT lines: These lines are vertical, i.e. parallel to the ordinate and are uniformly, spaced. These lines are drawn at an interval of 5°C and up to the saturation line.

Specific humidity or moisture content

lines: These lines are horizontal i.e parallel to the abscissa and are uniformly spaced.

DPT lines: These lines are horizontal, i.e. parallel to the abscissa and are non-uniformly spaced. At any point on the saturation line, the DBT and DPT are equal.

WBT lines: These are inclined straight lines and are non-uniformly spaced. At any point on the saturation line DBT and WBT are equal.

Enthalpy line: These are inclined straight lines and uniformly spaced. These lines are parallel to the WBT lines. The enthalpy values are given above the saturation line. Enthalpy

deviation

$(w_s - w)h_w$ is read form the deviation lines which are curved non-equally spaced with +ve and −ve correction.

Specific volume lines: These are obliquely inclined straight lines and are uniformly spaced.

Vapour pressure lines: These lines are horizontal and uniformly spaced. These lines are not drawn on the chart. A scale showing vapour pressure in mm of Hg is given on the extreme left or right side of the chart.

Relative humidity lines: These are curved lines and follow the saturated line. The saturation line represents 100% RH.

The psychrometric chart is shown in Fig. 20.6.

Figure 20.5 *Layout of psychrometric chart*

Figure 20.6 *Psychrometric chart*

20.8 ❐ PSYCHROMETRIC PROCESSES

The psychrometric processes by which the state of moist air can be altered are shown in Fig. 20.7. These processes are:

1. Sensible heating-process *OA*
2. Sensible cooling-process *OB*
3. Humidifying-process *OC*
4. Dehumidifying-process *OD*
5. Heating and humidifying-process *OE*
6. Cooling and dehumidifying-process *OF*
7. Cooling and humidifying-process *OG*
8. Heating and dehumidifying-process *OH*

Figure 20.7 *Basic psychrometric processes*

20.8.1 Sensible Heating or Cooling Process

The heating/cooling of air without any change in its specific humidity is known as sensible heating/cooling. The process

of sensible heating (line 1-2) or cooling (line 2-1) is shown by a horizontal line in Fig. 20.8. In this process, the specific humidity of air remains constant, i.e. $w_1 = w_2$. The heat has to be transferred which goes to change the temperature of air. The sensible heat transfer rate is given by,

$$\begin{aligned} \dot{Q}_s &= \dot{m}_a (h_2 - h_1) \\ &= \dot{m}_a [c_{pa} (t_2 - t_1) + wc_{pv} (t_2 - t_1)] \\ &= \dot{m}_a c_p (t_2 - t_1) \end{aligned}$$

where $c_p = c_{pa} + wc_{pv}$ = humid specific heat

$\dot{m}_a$ = mass flow rate of dry air, kg d.a./s

Figure 20.8 *Sensible heating process*

= Volume of air $\dot{V}$ in m^3/min × density of air ρ in kg/m^3

For the purpose of calculations, standard air is taken at 20°C and 50% RH. The density of standard air is approximately 1.2 kg/m^3 d.a. The value of humid specific heat is taken as 1.0216 kJ/(kg d.a.)(K)

The sensible heating of moist air can be done to any desired temperature. But the sensible cooling can be done only up to the dew point temperature t_{dp}, as shown in Fig. 20.8.

20.8.2 Humidification or Dehumidification Process

Humidification is the addition of moisture to air without change in its DBT. Similarly dehumidification is the removal of moisture without change in DBT. When the state of air is altered

along the constant DBT line, such as 1-2 in Fig. 20.9, moisture in the form of vapour has to be transferred to change the humidity ratio of air. This transfer of moisture is given by,

Because of this change in humidity ratio, there is also a change in the specific enthalpy of air, as shown in Fig. 20.9. This change in enthalpy due in the change in humidity ratio is considered to cause a latent heat transfer, given by

$$\dot{Q}_L = \dot{m}_a(h_2 - h_1) = \dot{m}_a[(c_p t_{d2} + h_{fg} w_2) - (c_p t_{d1} + h_{fg} w_1)]$$

Since $t_{d_1} = t_{d_2}$

Figure 20.9 *Humidification or dehumidification*

where h_{fg} = latent heat of vaporisation at t_{d1}

20.8.3 Heating and Humidification

This process involves both a change in temperature as well as in the humidity ratio as shown in Fig. 20.10. The change in temperature causes a sensible heat load, given by,

$$Q_s = \dot{m}_a (h_A - h_1) = \dot{m}_a c_p (t_{d2} - t_{d1})$$

The change in humidity ratio causes a moisture transfer, given by

$$G = m_a (w_2 - w_A) = m_a (w_2 - w_1)$$

Latent heat load $Q_L = m_a (h_2 - h_A) = m_a h_{fg} (w_2 - w_1)$

Figure 20.10 *Heating and humidification*

20.8.4 Sensible Heat Factor-SHF

The ratio of the sensible heat transfer to

the total heat transfer is termed as the sensible heat factor.

Thus

The process line 1-2 is called the sensible heat factor line (or process or condition line). The point *A* divides the total enthalpy change in the ratio of SHF and (1 – SHF). Q_s is proportional to SHF and Q_L to (1 – SHF).

20.8.5 Cooling and Dehumidification

In this process, the DBT as well as specific humidity of air decreases. The final relative humidity of the air is generally higher than that of the entering air. This is achieved by passing the air over a cooling coil or through a cold

water spray. The effective surface temperature of the coil is known as *apparatus dew point* (ADP). The cooling and dehumidification process is shown in Fig. 20.11.

$$Q_s = h_A - h_2$$
$$Q_L = h_1 - h_A$$
$$Q = Q_s + Q_L = h_1 - h_2$$

Figure 20.11 *Cooling and dehumidification*

20.8.6 Air Washer

The schematic representation of an air washer is shown in Fig. 20.12. It involves the flow of air through a spray of water. During the course of flow, the air may be cooled or heated, humidified or dehumidified, or simply adiabatically saturated, depending on the mean surface temperature of water. The water is, accordingly, externally cooled or

heated or simply circulated by a pump. Make-up water is added for any loss in the case of humidification of air. Eliminator plates are provided to minimise the loss of water droplets.

The thermodynamic changes of state of air along path 1-2 in an air washer, depending on the mean surface temperature of water droplets t_s (equal to t_u), is shown in Fig. 20.13. The following processes are possible:

1. **Process 1-2*a*:** Heating and humidification ($t_s > t_{d1}$). The water is externally heated.
2. **Process 1-2*b*:** Humidification ($t_s = t_{d1}$). The enthalpy of air increases. The water is externally heated.
3. **Process 1-2*c*:** Cooling and humidification ($t_{w1} < t_s < t_{d1}$). The mean surface temperature of water is greater than the wet bulb temperature of air, t_{w1}. The enthalpy of air increases as a result of humidification. The water has to be externally heated.
4. **Process 1-2*d*:** Adiabatic saturation ($t_{w1} = t_s$). The water is simply recirculated without any external heating or cooling.
5. **Process 1-2*e*:** Cooling and humidification ($t_{dp} < t_s < t_{w1}$). The enthalpy of air decreases. The water is required to be externally cooled.
6. **Process 1-2*f*:** Cooling ($t_s = t_{dp}$). Water is required to be cooled.
7. **Process 1-2*g*:** Cooling and humidification ($t_s < t_{dp}$). The

condition line drawn from the initial state 1 is tangential to the saturation line.

The mass balance of an air washer is shown in Fig. 20.14.

Figure 20.12 *Air-Washer*

Figure 20.13 *Range of psychrometric processes with an air washer*

Figure 20.14 *Mass balance of an air washer*

Let $\dot{m}_a$ = mass flow rate of dry air

$\dot{m}_w$ = mass flow rate of water

Energy balance gives,

$$\dot{m}_a (h_2 - h_1) = \dot{m}_w c_{pw} t_{w3} - [\dot{m}_w - \dot{m}_a (w_2 - w_1)] c_{pw} t_{w4}$$

or $\dot{m}_a (h_2 - h_1) = \dot{m}_w c_{pw} (t_{w3} - t_{w4}) t_{w3} + \dot{m}_a (w_2 - w_1) c_{pw} t_{w4}$

Neglecting the effect of temperature of water in the last term, we have

For adiabatic saturation case, $dh = 0$

$$d_{tw} = 0 \text{ and } t_{w3} = t_{w4}$$

20.8.7 Cooling with Adiabatic Humidification

When the air is passed through an insulated chamber, as shown in Fig. 20.15(a), having water sprays maintained at a temperature (t_f) higher than the DPT of entering air (t_{dp1}), but lower than its DBT (t_{d1}) of entering air or equal to the WBT of entering air (t_{w1}), then the air, is said to be cooled and humidified. Since the chamber is insulated and same water is circulated again and again, therefore, a condition of a adiabatic saturation is reached. Ultimately $t_f = t_{w1}$. This process is shown by line 1-3 in Fig. 20.15(b). This line follows the path along constant

WBT line or constant enthalpy line.

Figure 20.15 *Cooling with adiabatic humidification*

For perfect humidification, the final condition of air will be at point 3, where $t_{d3} = t_{w1}$. In actual practice the humidification is not perfect, therefore the final condition of air at outlet will be represented by point 2 on line 1-3.

Effectiveness or humidifying efficiency of spray chamber, for $t_f = t_{w1}$, is given by

When $t_f < t_{w1}$, then the process follows the path 1-2′-3′. In that case

When $t_f > t_{w1}$, then the process follows the line 1-2″-3″. In that case

20.8.8 Cooling and Humidification by Water Injection (Evaporative Cooling)

Let liquid water at temperature t_f be injected and sprayed into air stream with the help at nozzles. The condition of air will change depending on the amount of water that evaporates. The enthalpy of evaporation will come from the enthalpy of air. The process is shown in Fig. 20.16.

Let $\dot{m}_a$ = mass flow state of air

$\dot{m}_v$ = mass flow rate of water evaporated

= mass flow rate of water supplied.

h_f = enthalpy of liquid water

Then mass and enthalpy balances given

The term $(w_2 - w_1)\, h_f$ is extremely small as compared to h_1 and h_2.

Hence $h_1 = h_2$. Thus the water injection process is a constant enthalpy process, irrespective of the temperature of injected water.

Figure 20.16 *Cooling and humidification by water injection*

Figure 20.17 *Heating and humidification by steam injection*

20.8.9 Heating and Humidification by Steam Injection

Steam is normally injected into fresh outdoor air which is then supplied for the conditioning of textile mills where high humidity needs to be maintained. The process is shown in Fig. 20.17. If m_s is the mass of steam supplied with enthalpy h_s and m_a the mass of air, then

from the mass balance,

and for the enthalpy balance,

20.8.10 Heating and Adiabatic Chemical Dehumidification

This process is mainly used in industrial air-conditioning. It can also be used for some comfort air-conditioning requiring low relative humidity or low dew point temperature in the room. In this process, the air is passed over chemicals which have infinity for moisture. As the air comes in contact with these chemicals, the moisture gets condensed out of air and gives up its latent heat. Due to condensation, the specific humidity decreases and the heat of condensation supplies sensible heat for heating of air.

As a result of this the DBT increases. The process is shown in Fig. 20.18 by line 1-2. The path followed during the process is along the constant WBT line or constant enthalpy line.

The chemicals used are hygroscopic solutions or brines of calcium chloride, lithium chloride, lithium premie and ethyl glycol. These are called absorbents which undergo a chemical or physical or both changes during taking moisture. The other type of chemicals are called adsorbents which are in the solid state to take up moisture but do not undergo changes chemically or physically. These include silica gel and activated alumina. The effectiveness or efficiency of the dehumidifier,

Figure 20.18 *Heating and adiabatic chemical dehumidification*

20.9 ❐ ADIABATIC MIXING OF TWO AIR STREAMS

Consider two air streams 1 and 2 mixing adiabatically, as shown in Fig. 20.19(a)

Let m_{a1}, h_1, w_1 = mass, enthalpy and specific humidity of entering air at 1.

m_{a2}, h_2, w_2 = corresponding values of entering air at 2.

m_{a3}, h_3, w_3 = corresponding values of leaving air at 3.

The mixing process on psychrometic chart is shown in Fig. 20.19(b).

For the mass balance,

$$m_{a1} + m_{a2} = m_{a3}$$

For the energy balance,

$$m_{a1}h_1 + m_{a2}\, h_2 = m_{a3}\, h_3$$

For the mass balance of water vapour,

$$m_{a1}w_1 + m_{a2}\, w_2 = m_{a3}\, w_3$$

Eq. (20.42) can be written as,

Simplifying, we get

Figure 20.19 *(a) Adiabatic mixing of air streams, (b) Mixing process on the psychrometric chart*

The second term in the above expression being negligible, we get

Example 20.5

The atmospheric air at 760 mm of Hg, DBT = 20°C and WBT = 10°C

enters a heating coil whose temperature is 40°C. Assuming bypass factor of heating coil as 0.5, determine for the air leaving the coil: (*a*) DBT, (*b*) WBT, (*c*) relative humidity, and (*d*) sensible heat added to the air per kg of dry air.

Solution

Given: p_b = 760 mm Hg, t_{d1} = 20°C, t_{w1} = 10°C, t_{d3} = 40°C, BPF = 0.5

1. Let t_{d2} = DBT of air leaving the coil

   The psychrometric process is shown in Fig. 20.20.

   (b) and (c) from the psychrometric chart we find that t_{w2} = 14.4°C

   and RH, ϕ_2 = 15%

1. d. From the psychrometric chart,

$$h_1 = 30 \text{ kJ/kg. d.a.}$$

and h_2= 40 kJ/kg. d.a.

Figure 20.20

Sensible heat added to air = $h_2 - h_1$

$$= 40 - 30$$
$$= 10 \text{ kJ/kg.d.a.}$$

Example 20.6

Moist air enters a refrigeration coil at the rate of 120 kg d.a/min at 40°C and 50% RH. The apparatus dew point of coil is 10°C and by-pass factor is 0.20. Determine the outlet state of moist air and capacity of coil in TR.

Solution

Given: m_a =120 kg d.a/min, t_{d1} = 40°C, ϕ_1 = 50%, ADP = 10°C, BPF

$= 0.20$

1. Let t_{d2} and ϕ_2 be the DBT and RH of air leaving the cooling coil respectively

The psychrometric process is shown in Fig. 20.21.

Figure 20.21

Locate point 1 ($t_{d1} = 40°C$, $\phi_1 = 50\%$) on the psychrometric chart. Join point 1 with ADP = 10°C point on the saturation line.

2. Draw a vertical line at $t_{d2} = 16°C$ to intersect line 1-3 at point 2. Then $\phi_2 = 93\%$
3. From the psychrometric chart, $h_1 = 101$ kJ/kg d.a, and $h_2 = 43$ kJ/kg d.a.

Cooling capacity of coil, $Q = m_a (h_1 - h_2)$

Example 20.7

The atmospheric air at 30°C DBT and 70% RH enters a cooling coil at the rate of 220 m^3/min. The coil DPT = 15°C and its BPF = 0.15.

Determine

1. the temperature of air leaving the cooling coil,
2. the capacity of the cooling coil in *TR*,
3. the amount of water vapour removed per minute, and
4. sensitive heat factor for the process.

Solution

Given: t_{d1} = 30°C, ϕ_1 = 70%, V_1 = 220 m^3/min (cmm). ADP = t_{d3} = 15°C, BPF = 0.15

1. Let t_{d2} = temperature of air leaving the cooling coil

   The psychromatric process is shown in Fig. 20.22.

   At t_{d1} = 30°C and ϕ_1 = 70%, t_{dp1} = 24°C from psychrometric chart.

Figure 20.22

From psychrometric chart, we have

w_1 = 18.8 g/kg.d.a.
w_2 = 11.8 g/kg.d.a.
v_1 = 0.885 m^3/kg
h_1 = 78.5 kJ/kg.d.a.

$$h_2 = 47.8 \text{ kJ/kg.d.a.}$$
$$h_A = 61.8 \text{ kJ/kg.d.a.}$$

Mass of air flowing through cooling coil,

2. Capacity of cooling coil = $\dot{m}_a (h_1 - h_2) = 248.6(78.5 - 47.8)$
   $= 7632$ kJ/min
3. Amount of water vapour removed per minute,
$$\dot{m}_v = \dot{m}_a(w_1 - w_2)$$
$$= 248.6\ (18.8 - 11.8) \times 10^{-3} = 1.74 \text{ kg/min}$$
4. Sensible heat factor, SHF =

Example 20.8

In a certain environment, the DBT of air is 25°C and RH is 40%. Determine the specific humidity, dew point and wet bulb temperature of air. This air is cooled in an air washer using recirculated spray water and having a humidifying efficiency of 0.85. Find the DBT and DPT of air leaving the air washer.

Solution

Given: $t_{d1} = 25°C$, $\phi_1 = 40\%$, $\eta_H = 0.85$

The psychrometric process is shown in Fig. 20.23

At point 1 ($t_{d1} = 25°C$, $\phi_1 = 40\%$), $w_1 = 7.6$ g/kg d.a., $t_{dp1} = 10.4°C$

and $t_{w1} = 16°C = t_{d3}$

Let t_{d2} = DBT of air leaving the air washer

$$t_{d2} = 17.35°C$$

Then $t_{dp2} = 15.2°C$

Figure 20.23

Example 20.9

The atmospheric air at 25°C DBT and 15°C WBT is flowing at the rate of 120 cmm through a duct. The dry saturated steam at 100°C is injected into the air stream at the rate of 75 kg/h. Calculate the specific humidity, enthalpy, DBT, WBT, and RH of leaving air.

Solution

Given: t_{d1}=25°C, t_{w1} =15°C, V_1 =120 cmm, t_s = 100°C, m_s = 75 kg/h or 1.25 kg/min.

The psychrometric process is shown in Fig. 20.24

At state 1 (t_{d1} = 15°C, t_{w1} =15°C), w_1 = 6.6 g/kg d.a., v_1 = 0.853 m^3/kg, h_1 = 42.2 kJ/kg d.a.

Mass flow rate of air,

Figure 20.24

Enthalpy of saturated steam at 100°C from steam tables, h_s = 2676 kJ/kg

Locate point 2 (w_2 = 0.0155 kg/kg d.a., h_2 = 65.97 kJ/kg) on the psychrometric chart

Then t_{d2} = 260°C, WBT, t_{w2} = 22.4°C, ϕ_2 = 73%.

Example 20.10

Saturated air leaving the cooling section of an air-conditioning system at 15°C at the rate of 60 m^3/min is mixed adiabatically with outside air at 30°C and 60% RH at the rate of 20 m^3/min. Find the specific humidity, relative humidity, DBT and the volume flow rate of the mixture.

Solution

Given: $t_{d1} = 30°C$, $\phi_1 = 60\%$ $V_1 = 20$ m^3/min, $t_{d2} = 15°C$, $V_2 = 60$ m^3/min

The psychrometric process for mixing is shown in Fig. 20.25.

Locate point 1 ($t_{d1} = 30°C$, $\phi_1 =$ 60%) and point 2 ($t_{d2} = 15°C$ on the

saturation curve)

From the psychrometric chart, we have

$$h_1 = 71.6 \text{ kJ/kg.d.a.}, \; h_2 = 42.1 \text{ kJ/kg.d.a.}$$
$$v_1 = 0.88 \text{ m}^3\text{/kg.d.a.}, \; v_2 = 0.83 \text{ m}^3\text{/kg.d.a.}$$

Figure 20.25

$$0.3144(71.6 - h_3) = h_3 - 42.1$$
$$h_3 = 49.16 \text{kg/kg.d.a.}$$
$$w_3 = 0.012 \text{ kg/kg.d.a.}, \quad \phi_3 = 88\% \text{ and } t_{d3} = 18.8°\text{C},$$
$$v_3 = 0.843 \text{ m}^3\text{/kg.d.a.}$$
$$\dot{V}_3 = (m_{a1} + m_{a2})v_3 = (22.73 + 72.29) \times 0.843 = 80.1 \text{ m}^3\text{/min}$$

20.10 ❐ THERMAL ANALYSIS OF HUMAN BODY

The rate at which body produces heat is called the *metabolic* rate. The heat

produced by a normal healthy person while sleeping is called the *basal metabolic rate*, which is of the order of 60 W. The maximum value may be 10 times as much as this for a person engaged in sustained hard work. Human comfort is influenced by physiological factors determined by the rate of heat generation within the body and the rate of heat dissipation to the environment.

The body loses heat to the environment mainly by convection (Q_{conv}), radiation (Q_{rad}) and evaporation of moisture (Q_{evap}). In addition, there is heat loss by respiration having sensible component (Q_{Sr}) and latent component (Q_{Lr}) the total heat loss from the body is thus,

$$\begin{aligned} Q &= Q_{\text{conv}} + Q_{\text{rad}} + Q_{\text{evap}} + (Q_{Sr} + Q_{Lr}) \\ &= (Q_{\text{conv}} + Q_{\text{rad}} + Q_{Sr}) + (Q_{\text{evap}} + Q_{Lr}) \end{aligned}$$

$$= Q_{ST} + Q_{LT}$$

where Q_{ST} = total sensible heat loss

Q_{LT} = total latent heat loss.

The total sensible heat component depends on the temperature difference between the surface of the body and the surroundings. The latent heat component depends in the difference in the water vapour pressures.

In summer, the temperature difference available for sensible heat transfer is less. Thus, the convective and radiative heat losses are reduced. To maintain thermal equilibrium, the body starts perspiring to increase the evaporative loss. On the other hand in winter the sensible heat transfer is increased so that

the evaporative losses tend towards zero.

The heat exchange between human body and environment can be expressed by the following energy balance equation:

where Q_M = metabolic heat produced within the body

W= useful work done by human being

$Q_M - W$ = heat to be dissipated to environment

$Q = Q_{\text{conv}} + Q_{\text{rad}} + Q_{\text{evap}} + (Q_{Sr} + Q_{Lr})$

Q_s = heat stored in the body

1. The metabolic heat produced depends upon the rate of food energy consumption in the body.
2. The heat loss by evaporation is always positive. It depends

upon the vapour pressure difference between the skin surface and the surrounding air.

where C_d = diffusion coefficient, kg of water evaporated per unit surface area and pressure difference per hour.

A = skin surface area = 1.8 m^2 for a normal human being

p_s = saturation vapour pressure corresponding to skin temperature

p_v = vapour pressure of surrounding air

h_{fg} = latent heat of vaporisation = 2450 kJ/kg

C_c = factor which accounts for clothing worn.

3. The heat loss or gain by radiation from the body to the surroundings depends upon the mean radiant temperature. It is the average surface temperature of the surrounding objects. Q_{rad} is positive when the mean radiant temperature is lower than the DBT of room air, i.e. the human body will undergo a radiant heat loss. When Q_{rad} is negative, the body will undergo a radiant heat gain.
4. The heat loss by convection from the body to the surroundings is given by

where U = body film coefficient of heat transfer

A = body surface area = 1.8 m^2 for normal human being

t_b = body temperature

t_s = surroundings temperature.

The heat will be gained by body when $t_s > t_b$ and Q_{conv} is negative

5. Q_S is negative when $Q > (Q_M - W)$, i.e. body temperature falls down. Q_S is positive when

$Q < (Q_M - W)$, i.e. a human being gets fever.

A human body feels comfortable when there is no change in the body temperature, i.e. when $Q_S = 0$. Any variation in the body temperature acts as a stress on the brain which results in either perspiration or shivering.

20.10.1 Factors Affecting Human Comfort

The following factors affect human comfort:

1. Effective temperature.
2. Heat production and regulation in human body.
3. Heat and moisture losses from the human body.
4. Moisture content of air.
5. Quality and quantity of air.
6. Air velocity.
7. Hot and cold surfaces.
8. Air stratification.

20.10.2 Physiological Hazards Resulting from Heat

The physiological hazards resulting from rise in body temperature are:

1. **Heat exhaustion:** It is due to the failure of normal blood circulation. The symptoms of heat exhaustion include fatigue, headache, dizziness, vomiting and abnormal mental reactions such as irritation. Severe heat exhaustion may cause fainting. It does not cause permanent injury to the body and recovery is usually rapid when the person is shifted to a cool place.
2. **Heat Cramp:** It results from loss of salt due to an excessive rate of body perspiration. It causes severe pain in high muscles. This may be avoided by using salt tablets.
3. **Heat stroke:** It occurs when a person is exposed to excessive heat and work. If the body temperature rises rapidly to 40.5°C or 105°F, sweating ceases and a person may enter a coma, with imminent death. It may lead to permanent damage to brain. Heat stroke can be a avoided by drinking sufficient water at frequent intervals.

20.11 ❐ EFFECTIVE TEMPERATURE

Effective temperature (*ET*) is defined as the temperature of saturated air (RH = 100%) at which the human beings would experience the same feeling of comfort as experienced in the actual unsaturated environment. It may also be defined as that index which correlates the combined effects of air temperature,

relative humidity and air velocity on the human body.

Correspondingly, another effective temperature (ET*) can also be defined as the temperature at 50% RH at which the human body would experience exactly same feeling of comfort as at *ET* at 100% RH, and as experienced in the actual environment.

The effective temperature is defined to evaluate the combined effect of DBT, *RH*, and air velocity. The numerical value of effective temperature is made equal to the temperature of still (i.e. 5 to 8 m/min air velocity) saturated air, which produces the same sensation of warmth or coolness as produced under the given conditions.

20.11.1 Comfort Chart

The comfort chart is defined as that index which correlates the combined effects of air temperature, relative humidity and air velocity on the human body. The numerical value of effective temperature is made equal to the temperature of still (i.e. 5 to 8 m/min air velocity) saturated air, which produces the same sensation of warmth or coolness as produced under the given conditions.

The comfort chart, shown in Fig. 20.26, represents the concept of effective temperature in the comfort chart, DBT is taken as the abscissa and WBT as ordinates. The RH lines are replotted from the psychrometric chart. The

statistically prepared graphs corresponding to summer and winter season are also superimposed. These graphs have effective temperature scale as abscissa and percentage of people feeling comfortable as ordinate.

There is a correlation between comfort level, temperature, humidity sex, length of exposure, etc. According to ASHRAE (American Society for Heating, Refrigeration, and Air conditioning Engineering) thermal sensation scale for exposure for a period of one hour, the value of index y for comfort for men, women and both sexes separately are expressed as follows

where t_d = DBT, °C

p_v = vapour pressure, kPa

The values for feeling of comfort of *y* are as follows:

The comfort chart shown in Fig. 20.26 represent the range for both summer and winter conditions within which a condition of comfort exists for most people. The general practice is to recommend the following optimum inside design conditions for comfort:

Figure 20.26 *Comfort chart for still air (air velocities from 5 to 8 m/min)*

The comfort chart does not take into account the variations in comfort conditions when there are wide variations in the mean radiant

temperature. The comfort chart shown in Fig. 20.27 has become obsolete due to its shortcomings of over exaggeration of humidity at lower temperature and under estimation of humidity at heat tolerance level. The modified comfort chart shown in Fig. 20.27 is commonly used now.

Figure 20.27 *Modified comfort chart*

20.11.2 Factors Affecting Optimum Effective Temperature

The factors affecting optimum effective temperature are:

1. **Climate and seasonal differences:** The ET changes with season. During summer it is 22°C and during winter 19°C.
2. **Clothing:** Persons with light clothings need lower optimum ET than a person with heavy clothings.
3. **Age and sex:** Women require higher ET (about 0.5°C) than men. Children also need higher ET than adults.
4. **Duration of stay:** For shorter stay, higher ET is required than that needed for longer stay.
5. **Kind of activity:** Low ET is required for people doing heavy activity, like factory workers, dancers, etc.
6. **Density of occupants:** Lower ET is needed when density of

persons in a place is high.

20.12 ❒ SELECTION OF INSIDE AND OUTSIDE DESIGN CONDITIONS

20.12.1 Selection of Inside Design Conditions

The inside design conditions depend on the particular air-conditioning system. The important air-conditioning systems are:

1. Cold storage, 2. Industrial air-conditioning, 3. Comfort air conditioning

Table 20.1 *Storage conditions and properties of food products*

1. **Cold Storage:** In a cold storage air-conditioning system, the room air is cooled to much lower temperature over a cooling coil and supplied back to the storage space. The condition-maintained inside the storage space depend on the nature of the product stored. In cold storages, strict control of both

temperature and relative humidity is required. The required storage conditions for a number of products are given in Table 20.1.

2. **Industrial Air-conditioning:** The inside design conditions for industrial air-conditioning depend on the category of application. One category comprises those where constancy of temperature is the prime consideration, such as metrology laboratories, precision machine tools, computer centres, etc. In these applications a variation in relative humidity of 10 to 20 percent will not have much effect. The other category may comprise paper and textile mills where the relative humidity is to be maintained constant at a high value of 70 to 75 percent. The temperature requirements of such spaces are not severe. In another category of applications, strict control of both temperature and relative humidity is required, such as chemical and biological process industries.
3. **Comfort air-conditioning:** This has been dealt with in art 20.11.

20.12.2 Selection of Outside Design Conditions

For the outside design conditions in summer, it is recommended to use the mean monthly maximum dry bulb temperature and its corresponding wet bulb temperature.

During winter it has been observed that the fuel consumption for heating of buildings varies also directly as the

difference between the outside temperature and a reasonably comfortable inside temperature of 18.5°C. Thus the power consumption would be practically nil if the outside temperature is 18.5°C. On the other hand, the power consumption would double if the outside temperature drops from 18.5°C to 8.5°C. A degree-day is obtained for every degree when the mean outside temperature is below 18.5°C during the 24 hour period. If in a given locality the outside temperature average of 30 days is 10°C, then the degree days (*D*) for the period would be

$$D = (18.5 - 10) \times 30 = 255$$

The outside design temperature may be calculated as follows:

where t_{do} = outside design temperature.

20.13 ❒ COOLING LOAD ESTIMATION

The components of a cooling load are:

1. **Sensible heat gain:** This is the direct addition of heat to the space to be conditioned. It may occur due to any one or all of the following sources of heat transfer:
    1. The heat flowing into the building by conduction through exterior walls, floors, ceilings, doors and windows due to the temperature difference on their two sides.
    2. The heat received from solar radiation. It consists of the following:
        1. The heat transmitted directly through glass of windows, ventilators, and doors.
        2. The heat absorbed by walls and roofs exposed to solar radiation and later on transferred to the space to be conditioned by conduction.
    3. The heat conducted from adjoining un-conditioned rooms.
    4. The heat generated by fans, lights, machinery, cooking operations, industrial processes, etc.
    5. Occupancy load.
    6. Air infiltration load through doors, windows, and their frequent opening.
    7. Heat gain from walls of ducts, etc.
2. **Latent heat gain:** The latent heat load occurs due to the presence of water vapour in the air of conditioned space. This load occurs due to the following sources:
    1. Moisture in the air entering by infiltration
    2. Condensation of moisture from occupants, food cooking process.
    3. Moisture entering directly into the conditioned space through permeable walls, partitions, etc.

Total cooling load = Sensible heat load + Latent

heat load.

20.13.1 Heat Transfer Through Walls and Roofs

The heat transferred through a plane wall by conduction is given by Fourier law. For a single wall shown in Fig. 20.28(a), we have

where k = thermal conductivity of wall, W/(m. °C)

A_c = area of wall cross-section, m^2

Δt = temperature difference on two sides of wall, °C

Δx = wall thickness, m

The heat transfer by convection from a plane wall is given by Newton's law

$Q_{conv} = hA_s \, \Delta t$

where h = heat transfer coefficient, W/(m^2 °C)

A_s = surface area of wall, m^2

Δt = temperature difference between wall and surroundings, °C

Heat gained through a wall by the combined effect of conduction and convention becomes,

Figure 20.28 *Heat transfer by conduction through a plane wall: (a) Single wall, (b) Composite wall, (c) Composite wall with air space*

where subscripts o, i refer to outside and inside wall conditions respectively

and U = overall coefficient of heat

transfer of wall

for a single wall (Fig. 20.28a)

for a composite wall (Fig. 20.28b)

for a composite wall with air gap(Fig. 20.28c)

where k_a = thermal conductance of air space

20.13.2 Heat Gain from Solar Radiation

The heat from solar radiation is received by building surfaces by direct radiation and diffuse radiation. The direct radiation is the impingement of the Sun's rays upon the surface. The diffuse radiation is received from moisture and dust particles in atmosphere which

absorbs part of the energy of the Sun's rays. The diffuse radiation is received by surfaces which do not face the sun.

The heat gain through outside walls and roofs is given by,

where Δt_e = equivalent temperature differential.

20.13.3 Sol Air Temperature

It is a hypothetical temperature used to calculate the heat received by the outside surface of a building wall by the combined effect of convection and radiation.

Total heat received by the outside surface of the wall per unit area.

$$q_{os} = q_{\text{conv}} + q_{\text{rad}} = h_o\,(t_o - t_{os}) + I\,\alpha$$

where = sol air temperature.

t_o = temperature of outside air

t_{os} = temperature of outside surface of wall

h_o = outside film coefficient

I = total radiation intensity

α = absorptivity of the surface

20.13.4 Solar Heat Gain Through Glass Areas

20.13.5 Heat Gain Due to Infiltration

There are two methods of estimating the infiltrated air

(i) Crack length method, and (ii) Air change method.

The crack length method is usually used where greater accuracy is required. In most cases the air change method is used for calculating the quantity of infiltrated air. According to this method, the quantity of infiltrated air through windows and walls is,

where L, W, H = length, width and height of room respectively, m

A_c = air changes per hour

Factor is used because infiltration takes place on the windward side of building only.

20.13.6 Heat Gain from Products

This heat gain is very important in case of cold storages.

1. Chilling load above freezing,

   where m = mass of product

   c_{pm} = mean specific heat of product

   T_1, T_2 = initial and final temperature of product respectively

   t_{ch} = chilling time.

2.

   where h_{fg} = latent heat of freezing

   t_f = freezing time

3. Cooling load below freezing,

   where c'_{pm} = mean specific heat of freezing product

   $T_{1'}$ $T_{2'}$ = actual storage and freezing temperatures of product

   t_c = cooling time

4.

20.13.7 Heat Gain from Lights

Heat gain from lights, Q_t = total wattage of lights × use factor × allowance factor

= 0.5 for industrial workshops

Allowance factor = 1.25 for flourescent tubes.

20.13.8 Heat Gain from Power Equipments

where η_{em} = efficiency of electric motor.

20.13.9 Heat Gain Through Ducts

where U = overall heat transfer coefficient

A_D = surface area of duct

t_a, t_s = temperatures of ambient and supply airs respectively

20.13.10 Empirical Methods to Evaluate Heat Transfer Through Walls and Roofs

There are two approaches to calculate empirically heat transfer through walls and roofs.

They are:

1. The decrement factor and time lag method.
2. The equivalent temperature differential method.

Both the methods use analytical-experimental results for their formulations. The equivalent temperature differential method is more commonly used by the air-conditioning engineers as it is also applicable to sunlit walls and roofs.

1. **Using Decrement Factor and Time Lag:** If the thermal capacity of the wall is ignored, then the instantaneous rate of heat transfer through the wall at any time τ is given by

$$Q_\tau = UA(t_e - t_i)$$

where t_e = sol-air temperature.

and on an average basis, the mean heat flow is given by

$$Qm = UA(tem - ti)$$

where t_{em} = mean sol-air temperature.

For the sake of simplicity, the *dot* above Q has been dropped to indicate the *rate*.

But most building materials have a finite thermal capacity which is expressed as

$$mc = \rho cV = \rho c(A\Delta x)$$

where m = Mass of wall, V = volume of wall

ρ, c = Density and specific heat of wall material

A = Cross-sectional area of wall

Δx = Wall thickness.

It has been seen that there is a two-fold effect of thermal capacity on heat transfer

1. There is a time lag between the heat transfer at the outside surface q_o and the heat transfer at the inside surface q_i, defined by ϕ.
2. There is a decrement in the heat transfer due to the absorption of heat by the wall and subsequent transfer of a part of this heat back to the outside air when its temperature is lower, defined by λ.

The rigorous analytical method to determine the

time lag ϕ and decrement factor λ is quite complicated. In the limiting case, when the wall thickness approaches zero $\lambda \to 1$ and $\phi \to 0$

Then $Q_\tau = UA(t_e - t_i)$

i.e. the heat transfer through the wall is equal to its instantaneous value.

Considering the effect of thermal capacity, the actual heat transfer at any time τ is,

$$Q_\tau = UA(t_{em} - t_i) + UA\,\lambda(t_{e^{\tau-\phi}} - t_{em})$$

where $t_{e^{\tau-\phi}}$ = sol-air temperature at time $\tau - \phi$. i.e., ϕ hours before the heat transfer is to be calculated.

In a locality where the daily range of variation of the outside air temperature is small it is immaterial what thickness of wall is provided. But in a locality when the daily range of temperature is large, it is desirable to have thick walls so as to cut the cooling load in summer and the heating load in winter. Such walls also do not allow the inside temperature to rise very much during the day and drop at night.

2. **Equivalent temperature differential (*ETD*) method:** The actual heat transfer can be written in terms of an equivalent temperature differential Δt_E defined by the equation.

$$Q = UA\Delta t_E$$

where $\Delta t_E = (t_{em} - t_i) + \lambda(t_{e\tau-\phi} - t_{em})$

Δt_E depends on the following:

1. l and f, which in turn depend on the thermo-physical properties of construction.
2. Outside air temperature t_0 and solar radiation intensity I.
3. Room temperature t_i.

Thus, the equivalent temperature differential approach takes care of the exposure of the wall or roof to the sun.

20.14 ❒ HEATING LOAD ESTIMATION

An estimate of the heating load is made on the basis of maximum probable heat loss of the conditioned heated space. The heating loads are:

1. **Transmission heat loss**: The transmission heat loss from walls roofs, etc., is given by,

   where t_i, t_o = inside and outside design temperature respectively

2. **Solar radiation**: The solar radiation heat gain is neglected as peak heating load occurs in the early hours of morning.
3. **Internal heat gains**: The internal heat gains from occupants, lights, appliances, etc., decrease the heating load. These negative loads should be accounted for in applications, such as theatres, assembly halls, stores, office buildings, etc.,

20.15 ❐ ROOM SENSIBLE HEAT FACTOR (RSHF)

It is defined as the ratio of room sensible heat to the total heat (Fig. 20.29).

where RSH, RLH, RTH = room sensible, latent and total heats respectively.

Figure 20.29 *Room and supply air conditions and ADP*

Figure 20.30 *Grand sensible heat factor*

20.15.1 Estimation of Supply Air Conditions

When the supply air conditions are not known, then the RSHF line may be drawn from the RSHF calculated value, as described below (Fig. 20.30).

1. Mark point '*a*' on the sensible heat factor scale drawn on the right hand corner of psychrometric chart. The point '*a*' represents the calculated value of RSHF.
2. Joint point '*a*' with the alignment circle or reference point *b* (t_d – 26°C, ϕ = 50%). The line *ab* is called the base line.
3. Mark point *R* on the psychrometric chart to represent the room design conditions.

4. Draw a line *RR'* through point *R* parallel to the base line *ab*. This line is required *RSHF* line.

20.16 ❐ GRAND SENSIBLE HEAT FACTOR

It is defined as the ratio of total sensible heat to the grand total heat which the cooling coil or conditioning apparatus is required to handle (Fig. 20.31).

where TSH = total sensible heat = RSH + OASH

TLH = total latent heat = RLH + OALH

GTH = grand total heat = TSH + TLH = (RSH + OASH) + (RLH + OALH)

Let $\dot{V}_1$ = volume rare of outside air m^3/min

t_{d1}, t_{d2} = *DBT* of outside and room air

respectively, °C

w_1, w_2 = specific humidity of outside and room air respectively kg/kg.d.a

h_1, h_2 = enthalpy of outside and room air respectively, kJ/kg d.a

Outside air total heat, OATH = *OASH* + *OALH*

Figure 20.31 *RSHF line estimation*

20.17 ❐ EFFECTIVE ROOM SENSIBLE HEAT FACTOR

It is defined as the ratio of the effective room sensible heat to the effective room total heat

where ERSH = Effective room sensible heat

$$= \text{RSH} + \text{OASH} \times \text{BPF}$$

$$= \text{RSH} + 0.0204\ \dot{V}_1(t_{d1} - t_{d2}) \times \text{BPT}$$

Figure 20.32 *Effective room sensible heat factor*

ERLH = Effective room latent heat

$$= \text{RLH} + \text{OALH} \times \text{BPF}$$

$$= \text{RLH} + 50\ \dot{V}_1(w_1 - w_2) \times \text{BPF}$$

ERTH = Effective room total heat

$$= \text{ERSH} + \text{ERLH}$$

BPF = By-pass factor

The line joining the point 2 and point 6, i.e. *ADP* as shown in Fig. 20.32, gives the effective room sensible heat factor line (ERSHF line). From point 4, draw

line 4-4′ parallel to line 3-2. From similar triangles 6-4-4′ and 6-3-2, we have

20.18 ❐ AIR CONDITIONING SYSTEMS

20.18.1 Summer Air-conditioning System with Ventilation Air and Zero By-pass Factor

The schematic arrangement of summer air-conditioning system with ventilation without by-pass factor is shown in Fig. 20.33. The outside air flows through the damper and mixes with the recirculated air. The mixed air then passes through the filter to remove dirt and dust and then through cooling coil whose temperature is kept much below the DBT of air in the conditioned space. The cooled air then passes through a

perforated membrane where it losses its moisture in the condensed form. The moisture is collected in a sump. After that, the air is made to pass through a heating coil which heats the air slightly as per the requirement of designed air DBT and RH. Now the conditioned air is supplied to the room by a supply air fan. From the room a part of the used air is exhausted to the atmosphere by ventilation. The remaining part of the used air (recirculated air) is mixed with fresh air for conditioning.

Figure 20.33 *Schematic diagram of system with ventilation air without bypass factor*

Figure 20.34 *Summer air conditioning process with ventilation air without BPF*

Figure 20.34 shows the air-conditioning processes.

o, i = outside and inside air states respectively

1 = state of air after mixing recirculated room air with ventilation air

$\dot{m}_{ai}$, $\dot{m}_{a0}$ = mass flow rate of recirculated room air and ventilation air respectively

2 = supply air state for a minimum rate of supply air

$$\text{line } i\text{-2} = ADP,\ t_{dp2}$$
$$= RSHF \text{ line}$$
$$\text{line 1-2} = GSHF \text{ line}$$
$$\dot{m}_{as} = \text{mass flow rate of mixture air}$$
$$\dot{Q} = \dot{m}_{as}(h_i - h_2) + (\dot{m}_{ai}\, h_i + \dot{m}_{ao}\, h_\mathrm{o})$$
$$(\dot{m}_{as} - \dot{m}_{ao})h_i + \dot{m}_{ao}\, h_\mathrm{o} - \dot{m}_{as}\, h_2$$
$$= \mathrm{RTH} + \mathrm{OATH}$$

1. **Ventilation load**

   Let $(\mathrm{cmm})_o$ = outside ventilation air volume flow rate $= \dot{m}_{as} \times v_o$

2. **Room load**

Let $(cmm)_1$ = mixture air volume flow rate

$$\dot{m}_{a1} = \dot{m}_{ai} + \dot{m}_{ao} = \dot{m}_{as}$$

$$RTH = RSH + RLH$$

3. **Air-conditioning equipment load**

$$TSH = RSH + OASH$$
$$TLH = RLH + OALH$$
$$GTH = TSH + TLH$$

20.18.2 Summer Air-conditioning System with Ventilation Air and By-pass Factor

The summer air-conditioning processes with ventilation air and BPF is shown Fig. 20.35.

Figure 20.35 *Summer air-conditioning processes with ventilation air and BPF*

20.18.3 Winter Air-conditioning System

The schematic arrangement of winter air-conditioning system is shown in Fig. 20.36. The outside air flows through a damper and mixes up with the

recirculated air. The mixed air passes through a filter to remove dirt, dust and other impurities. The air now passes through a preheating coil to prevent the possible freezing of water and to control the evaporation of water in the humidifier. Then the air is passed through a reheating coil to bring the air to the designed DBT. This conditioned air is supplied to the room by means of a supply an fan. From the room, a part of the used air is exhausted to the atmosphere and the remaining part is recirculated again.

Figure 20.36 *Winter air-conditioning system*

The air-conditioning processes are shown in Fig. 20.37.

Process 1-2: Preheating

Process 2-3: Adiabatic saturation.

Process 1-3: Heating and humidifying air in air washer with pumped recirculation and external heating of water.

Process 3-*s*: Reheating

Figure 20.37 *Winter air-conditioning processes*

Example 20.11

Air at 10°C DBT and 90% RH is to be brought to 35°C DBT and 22.5°C WBT with the help of winter air-conditioner. If the humidified air comes out of the humidifier at 90% RH, determine: (a) the temperature to which the air should be

preheated, and (b) the efficiency of the air washer.

Solution

Given: t_{d1} =10°C, ϕ_1 = 90%, t_{ds} = 35°C, t_{ws} = 22.5°C

The psychrometric processes are shown in Fig. 20.38

Locate point 1(t_{d1}, ϕ_1) and point s(t_{ds}, t_{ws}).

Draw horizontal lines from points 1 and s. From point 3 draw WBT = const. line to locate point 2.

From psychrometric chart, the temperature to which air should be

preheated

$$t_{d_2} = 31.2°C$$
$$t_{d_3} = 18.5°C \text{ and } t_{d_3'} = 17.5°C$$

Efficiency of air-washer =

Figure 20.38

20.18.4 Comfort Air-conditioning System

In comfort air-conditioning, the air is brought to the required DBT and RH for the human health, comfort and efficiency. The comfort conditions have been discussed in detail in art. 20.11.1. Normally it is assumed to be 21°C DBT and 50% RH. The SHF is generally kept between 0.7 to 0.9. The comfort air-conditioning may be adopted for homes, offices, shops, restaurants, theatres, hospitals, schools, etc.

20.18.5 Industrial Air-conditioning System

In an industry the inside DBT and *RH* of air have to be kept constant for proper working of the machines and manufacturing processes. Special type of air-conditioning systems have to be provided for textile mills, paper mills, machine-parts manufacturing plants, tool rooms, CNC machining centres, photo-processing plants, etc.

Example 20.12

Atmospheric air at a pressure of 1-0132 bar has a dry bulb temperature of 30°C and Relative Humidity of 70%. Calculate:

1. partial pressure of Water Vapour and air in moist air,
2. the specific humidity,
3. the degree of saturation, and

4. the Dew Point temperature.

Solution

Given: $p_b = 1.0132$ bar, $t_d = 30°C$, $\phi = 70\%$

1. Corresponding to $t_d = 30°C$, saturation pressure of water vapour, from steam tables, is

$$p_s = 0.04246 \text{ bar},$$

Now $p_b = p_a + p_v$

Partial pressure of air, $p_a = 1.0132 - 0.02972 = 0.98348$ bar

2. Specific humidity,
3. Specific humidity of started air:

Degree of saturation,

4. DPT = saturation temperature corresponding to $p_v = 0.02972$ bar

From steam tables, corresponding to $p_v = 0.02972$ bar

Example 20.13

In a laboratory test on a particular day, a psychrometer recorded the dry bulb and wet bulb temperatures

for atmospheric air as 35°C and 25°C respectively. The atmospheric pressure is 1.013 bar 1. Calculate:

1. the relative humidity,
2. the specific humidity,
3. the dew point temperature, and
4. the degree of saturation.

The partial pressure of water vapour (p_v) can be calculated with the help of Carrier's Equation, given as;

where p_{wb} = Saturation pressure corresponding to wet bulb temperature,

p = Barometric pressure, and

t_{db} and t_{wb} = are the dry bulb and wet bulb temperatures of air (in 0°C).

Gas constant of air, $R_a = 0.287$ kJ/kg.K.

Solution

Given: $t_d = 35°C$, $t_w = 25°C$, $p_b = 1.013$ bar,

1. Corresponding to $t_d = 35°C$, $p_s = 0.5622$ bar, and for $t_w = 25°C$, $p_w = 0.03166$ bar

   Partial pressure of water vapour,

   Relative humidity,

2. Specific humidity,
3. Corresponding to $p_v = 0.02516$ bar,
4. Specific humidity of saturated,

   Degree of saturation,

Review Questions

1. Explain the following terms:
   1. Degree of saturation
   2. Specific humidity
   3. Relative humidity
   4. Dew point temperature
   5. Thermodynamic wet bulb temperature

2. Represent the following processes on psychrometric chart:
    1. Sensible heating and cooling
    2. Adiabatic heating and dehumidification
    3. Adiabatic cooling and humidification
    4. Cooling and dehumidification
3. What is the difference between WBT and thermodynamic WBT?
4. Define SHF and LHF.
5. What is by-pass factor?
6. What are the fa ctors that determine human comfort?
7. What is effective temperature? List the factors that affect effective temperature.
8. Differentiate between heat stroke and heat camp.
9. What is the difference between summer and winter air conditioning?
10. What is alignment circle?

Exercises

20.1. An air conditioned space is to be supplied with 300 cmm of air at 20°C dry bulb and 40% relative humidity. Unconditioned air from atmosphere at 32°C dry bulb and 28°C dew point is supplied to the dehumidifier coils, from where the same passes on to a heating coil before entering the conditioned space. Determine the cooling required. Solve the problem both from first

principles and with Psychrometric chart.

20.2. Room air at 26°C dry bulb and 50% relative humidity is mixed with outdoor air at 43°C dry bulb and 40% relative humidity in the ratio of 4 : 1. The mixture is passed through a cooling coil maintained at 5°C with a by-pass factor of 0.15. The air from the cooling coil is mixed with room air in the ratio of 4 : 1. The mixture is then reheated to 20°C dry bulb and supplied to the conditioned space.

1. Show the flow diagram schematically.
2. Show the different process on a Psychrometric chart.
3. For 450 kg of supply air per minute determine the quantity of fresh air needed, the refrigeration load and the heat supplied to the reheater coils.

20.3.

Determine:

1. effective sensible heat factor
2. dehumidified air-quantity
3. supply air temperature

Assume by-pass factor = 0.05.

20.4. In an Industrial Application:

Use all outdoor air and evaporative cooling process with washer efficiency 80%. Determine

1. Room DBT maintained
2. Supply quantity in cmm.

20.5. The following data refer to a laboratory

Also all out side fresh air to be supplied through coil, and no return air.

Determine

1. effective sensible heat factor
2. ADP
3. actual quantity of outdoor air to be supplied to take care of the loads
4. dehumidified air quantity supplied.

Assume by-pass factor = 0.05.

20.6. Air 35°C dry bulb temperature and 25°C wet bulb temperature is passed through a cooling coil at the rate of 280 m^3/min. The air leaves the cooling coil at 26.5°C dry bulb temperature and 50% relative humidity. Find

1. Capacity of the cooling coil in tonnes of refrigeration;
2. Wet bulb temperature of the leaving air;
3. Water vapour removed per minute: and
4. Sensible heat factor.

[**Ans.** 31.98 *TR*; 19.2°C; 1.56 kg/kg of dry air:0.39]

20.7. The outside air at 30°C dry bulb temperature and 18°C wet bulb temperature enters a cooling coil at the rate of 40 m^3/min. The effective surface

temperature of the cooling coil is 4.5°C and its cooling capacity is 12.5 kW of refrigeration. Find:

1. dry bulb and wet bulb temperatures of the air leaving the coil,
2. enthalpy of air leaving the coil, and
3. by-pass factor of the coil.

[**Ans.** 18.8°C; 12.7°C; 35.6 kJ/kg of dry air; 0.52]

20.8. Air at 40°C dry bulb temperature and 15% relative humidity is passed through the adiabatic humidifier at the rate 200 m^3/min. The outlet conditions of air are 25°C dry bulb temperature and 20°C wet bulb temperature. Find (a) dew point temperature, (b) relative humidity of exit air; and (c) amount of water vapour added to the air per minute.

[**Ans.** 17.8°C; 65%; 1 26 kg/min]

20.9. Air at 0°C and 95% relative

humidity has to be heated and humidified to 25°C and 40% relative humidity by the following three processes:

1. preheating
2. adiabatic saturation in a recirculated water air washer and
3. reheating to final state.

Calculate:

1. The heating required in two heaters
2. The makeup water required in washer and temperature of washer

   Assume effectiveness of washer as 80 per cent.

[**Ans.** 27.3 kJ/kg of dry air; 9 kJ/kg of dry air; 4.1 g/kg of dry air 12.6°C]

20.10. The following data relates to a cold storage for storing 450 tonnes of vegetables

If the air conditioning is achieved by first cooling and dehumidifying and then

heating and the temperature of air entering the room is not to exceed 16°C, determine

1. amount of recirculated air, if the recirculated air is mixed with fresh air before entering the cooling coil.
2. capacity of the cooling coil in tonnes of refrigeration and its by-pass factor if the dew point temperature of the coil is 6°C and
3. capacity of heating coil

[**Ans.** 3.384 kg/s; 46.75 TR; 34 82 kW]

20.11. The following data relates to a conference room having a seating capacity of 80 persons.

If two third of recirculated room air and one third of fresh air are mixed before entering the cooling coils determine

1. apparatus Dew point
2. grand total heat load
3. effective room sensible heat factor.

20.12. A sling psychrometer reads 40°C

DBT and 28°C WBT. Calculate the following:

1. Specific humidity
2. Relative humidity
3. Dew point, temperature
4. vapour density
5. Enthalpy of mixture per kg of dry air.

Assume atmospheric pressure = 1.03 bar

[**Ans.** 0.0189 kg/kg d.a., 40.4%, 24°C, 0.0338 kg/m^3, 88.8 k/kg d.a]

20.13. The atmospheric conditions are 25°C DBT and specific humidity of 10 g/kg of d.a. Determine the partial pressure of vapour and relative humidity.

Take atmospheric pressure = 1 bar.

[**Ans.** 0.0158 bar 49%]

20.14. Air enters an evaporative cooler at a pressure of 1 atm. DBT of 38°C and 25% RH. The air leaves the cooler at 1

atm and 18°C DBT. Assuming adiabatic process, determine the relative humidity of cool air and quality of water that must be supplied to cooler for each kg of dry air.

[**Ans.** 63%, 0.00808 kg/kg. d.a]

Appendix A

A.1 ❒ STEAM TABLES*

Table A.1.1 *Saturates steam: Temperature table*

* Adapted from Joseph H. Keenan, Frederick G.Keyes, Philip G. Hill, and Joan G. Moore, *Steam Tables*, John Wiley and Sons, New York, 1969.

Table A.1.2 *Saturated water: Pressure table*

Table A.1.3 *Superheated vapour*

Table A.1.4 *Compressed liquid water*

A.2 ❒ THERMODYNAMIC PROPERTIES OF REFRIGERANT-134A (1,1,1,1–TETRAFLUOROETHANE)

Table A.2.1 *Saturated R-134a*

Table A.2.2 *Superheated R-134a*

A.3 ❒ THERMODYNAMIC PROPERTIES OF AMMONIA

Table A.3.1 *Saturated ammonia table*

Table A.3.2 *Superheated ammonia table*

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